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CR-134873

QUIET CLEAN SHORT-HAUL EXPERIMENTAL ENGINE (QCSEE)

Ball Spline Pitch Change Mechanism Design Report

APRIL 1978

by

Advanced Engineering and Technology Programs Department

GENERAL ELECTRIC COMPANY

(NASA-CR-134873) QUIET CLEAN SHORT-HAUL EXPERIMENTAL ENGINE (QCSEE). BALL SPLINE PITCH CHANGE MECHANISM DESIGN REPORT (General Electric Co.) 73 p HC A04/MF A01 CSCL 21E G3/07

N80-15101

Unclas 33481

Prepared For

National Aeronautics and Space Administration

Contract No. NAS3-18021

1. Report No. NASA CR-134873	2. Government Accession No.	3. Recipient's Catalog No.

4. Title and Subtitle

QUIET CLEAN SHORT-HAUL EXPERIMENTAL ENGINE (QCSEE)
BALL SPLINE PITCH CHANGE MECHANISM DESIGN REPORT

5. Report Date

April 1978

6. Performing Organization Code

7. Author(s)

Advanced Engineering and Technology Programs Department
Group Engineering Division

8. Performing Organization Report No.

R77AEG327

9. Performing Organization Name and Address

General Electric Company
Cincinnati, Ohio 45215

10. Work Unit No.

11. Contract or Grant No.

NAS3-18021

12. Sponsoring Agency Name and Address

National Aeronautics and Space Administration
Washington, D.C. 20546

13. Type of Report and Period Covered

Contractor Report

14. Sponsoring Agency Code

15. Supplementary Notes

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16. Abstract

The QCSEE Program provides for the design of a variable-pitch change mechanism. Detailed design parameters are presented. A mechanical system containing a ball screw/spline driving two counteracting master bevel gears meshing pinion gears attached to each of 18 fan blades is provided. A design analysis has been completed for this system.

17. Key Words (Suggested by Author(s))

Mechanical Actuator
Pitch Change Mechanism

19. Security Classif. (of this report) Unclassified	20. Security Classif. (of this page) Unclassified		

* For sale by the National Technical Information Service, Springfield, Virginia 22151

NASA-C-168 (Rev. 6-71)

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1.0 SUMMARY

The ball spline actuation system shown in Figure 1 is being developed by General Electric. A hydraulic motor located on the fan centerline drives a ball screw actuator through a differential gear and no-back. Linear motion of the ball nut of the ball screw causes the translating sleeve (middle member) of a ball spline to move in a fore or aft direction. The ball spline is a double-acting member with helical ball tracks between the translating sleeve and inner member and straight ball tracks between the sleeve and the outer ball spline member. The inner member is attached to the aft ring gear while the outer member is attached to the forward ring gear. Translation of the ball spline sleeve fore and aft drives the two ring gears in tangentially opposite directions. The ring gears, in turn, are mated to 18 pinion gears that are splined to the corresponding fan blade trunnion.

Gear ratio between the hydraulic motor and the fan blade is 479/1. Two LVDT's driven by the hydraulic motor provide a blade angle feedback to the engine digital control system.

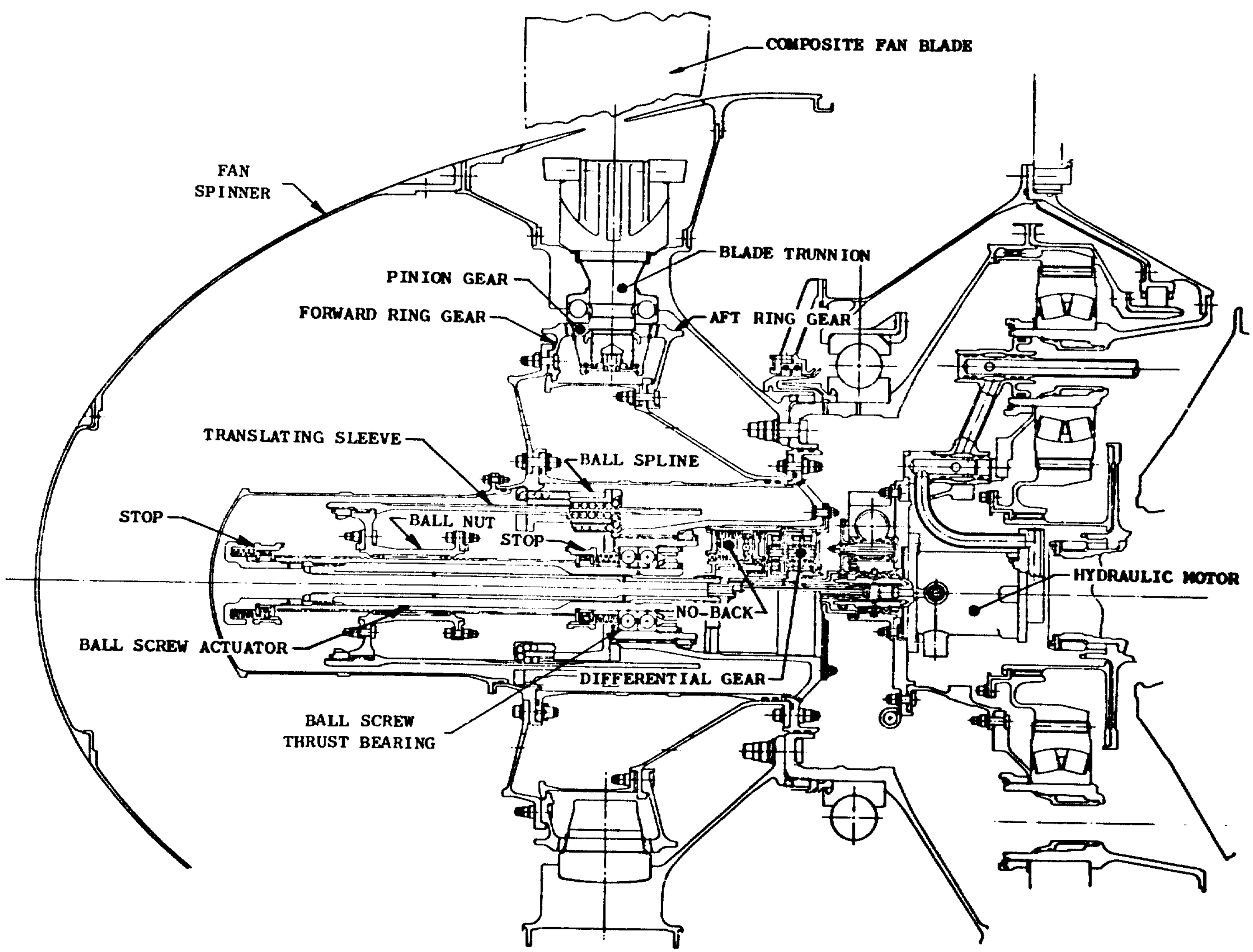


Figure 1. General Electric Ball Spline Actuation System.

2.0 INTRODUCTION

The Quiet Clean Short-Haul Experimental Engine (QCSEE) Program provides for the design, fabrication, and testing of experimental, high-bypass, geared turbofan engines and propulsion systems for short-haul passenger aircraft. The under-the-wing (UTW) propulsion system, being developed as part of this program, is a low tip speed, low pressure ratio turbofan engine that features a variable-pitch fan. Details of the propulsion system design are provided in Reference 1.

A tradeoff study was made comparing various pitch change mechanism concepts. These concepts included a ball spline, planetary gear and "mini" gear actuator. A summary of these concepts and the results of the study are given in Appendix A. As a result of this study the ball spline actuator was selected for engine testing in the UTW experimental engine program.

A second concept featuring a cam/harmonic drive system being developed by the Hamilton Standard Division of United Technology Corporation under subcontract to the General Electric Company will also be engine tested. This actuation system is described in Reference 2.

An oral review of the General Electric design was conducted for NASA representatives on February 25, 1975 at the General Electric plant in Evendale, (Cincinnati) Ohio. Design approval by NASA, and authorization to proceed with preparation of detailed drawings, finalization of subcontracts, and procurement and fabrication of components was received shortly thereafter.

This report presents final design and analysis results for the General Electric variable-pitch fan actuation system.

3.0 DESIGN REQUIREMENTS AND CRITERIA

The General Electric variable-pitch fan systems are designed to meet specified flight engine and experimental engine requirements, including resistance to bird strikes. These requirements are defined in detail in GE Specification M50TF1635-S1. A life requirement of 36,000 hours (48,000 mission cycles) was specified for all component parts, with the exception of bearings and standard nonreusable parts, when the actuation system was operated in accordance with the conditions specified in the Mission Duty Cycle presented in Figure 2. The mechanisms were designed for no replacement of parts (including bearings and nonreusable parts) at intervals of less than 9000 hours.

Both variable-pitch systems, including retention bearings, also must meet the duty cycle requirements of the experimental UTW engine, defined as follows:

EXPERIMENTAL ENGINE DUTY CYCLE

Item	Fan Speed, (rpm)	Time, (hours)
1	3447	0.3
2	3347	1
3	3157	181
4	2841	500
5	2368	1000
6	947	1000

The above operation is assumed to occur on a 90° F day. The actuation system must be capable of operation at any of the above conditions. For the experimental engines, the system operation capability also should be consistent with the following:

- Item 1
  - No actuation required
- Item 2
  - 10 actuations at 5-minute intervals in the forward range of operation from 5° open to 5° closed.
- Item 3
  - 500 total reversals through flat pitch and stall pitch
  - 1000 actuations at 5-minute intervals in the forward range of operation from 5° open to 10° closed

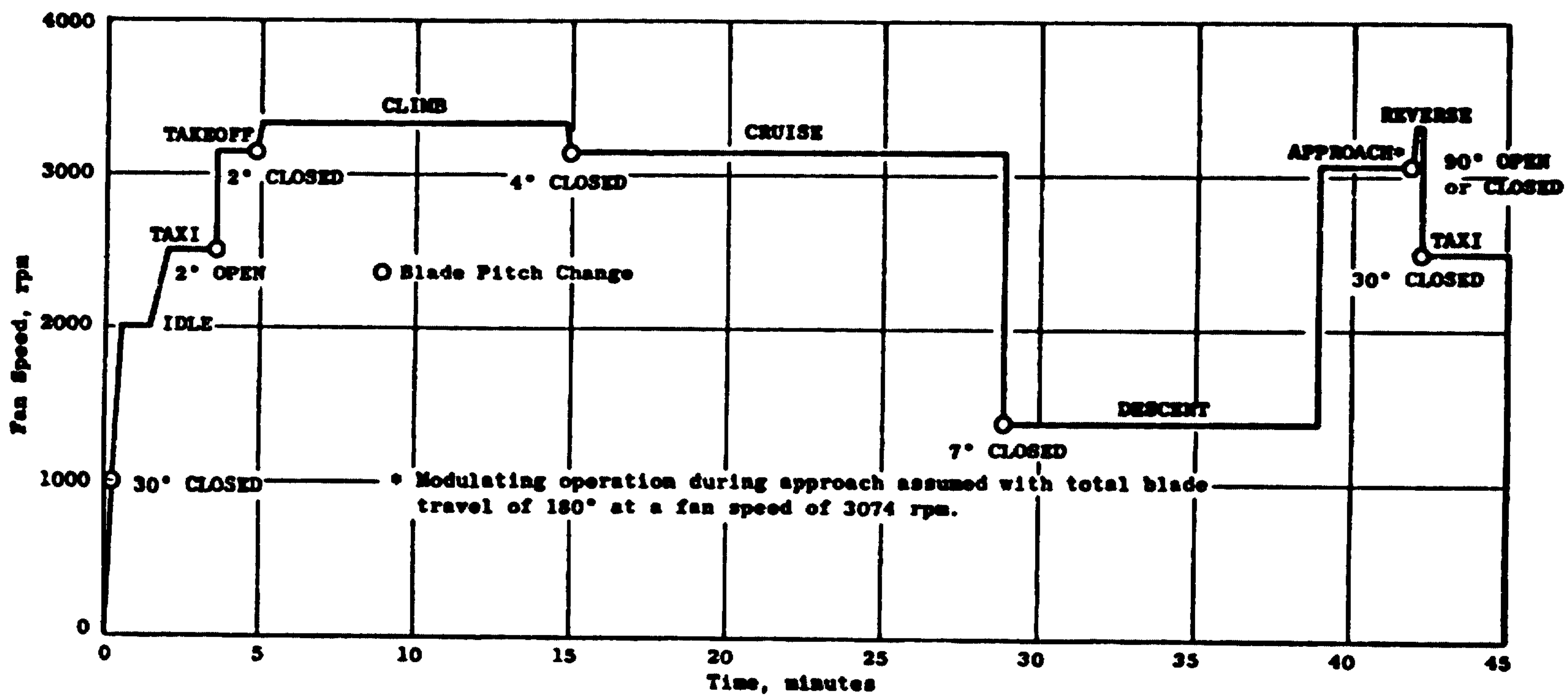


Figure 2. UTW Mission Duty Cycle.

- Item 4
  - 10,000 actuations in 5° increments from 5° open to 30° closed.

Specified blade-torque, speed, and accuracy design requirements are shown in Table I.

Figure 3 is a plot of the expected blade twisting moments in the operating range. The designs also are based on consideration of the following aircraft flight requirements;

- Following foreign object ingestion, the dynamic pitch change mechanism must be capable of 10 additional cycles at maximum torque and actuation rate.

- The system must be capable of withstanding 20 g's vibratory and should not impose over 5 g's vibratory on the engine.

- The system must be capable of operation after exposure to fluids conventionally used by Airlines such as "Skydrol Type" hydraulic fluids, methylene chloride, and butyl cellasolve.

Table I. Variable-Pitch System Design Requirements.

	Normal Range	Extreme Range
Fan Speed, rpm	0 - 3326	3450 without Actuation
Blade Twisting Torque[1]	See Figure 3	1692 N-m (15,000 lb-in.)[2]
Blade Overturning Moments		22,600 N-m (200,000 lb-in.)[2]
Centrifugal Load for Blade and Dovetail Only	15,196 kg (33,500 lb) (at 3200 rpm)	Must Not Burst at 4432 rpm (141% of T/O rpm)
Average actuation Rate	135°/sec	
Actuation Jogging at Blades	0.5° Steps Min.	
Feedback Signal Accuracy	±0.25° Blade Position	
Flight Maneuver Forces[3]	Per Mil-E-5007C (12/30/65) Par. 3.14 Except Precession Rate Shall be One Radian per sec Max.	

(1) Friction must be included over and above these data

(2) FAA Advisory Circular AC 33-1b, 4/22/70 (Bird Strike Torque)

(3) Polar moment of inertia of blades = 1273kN-cm^2 (44,376 lb-in.^{2})

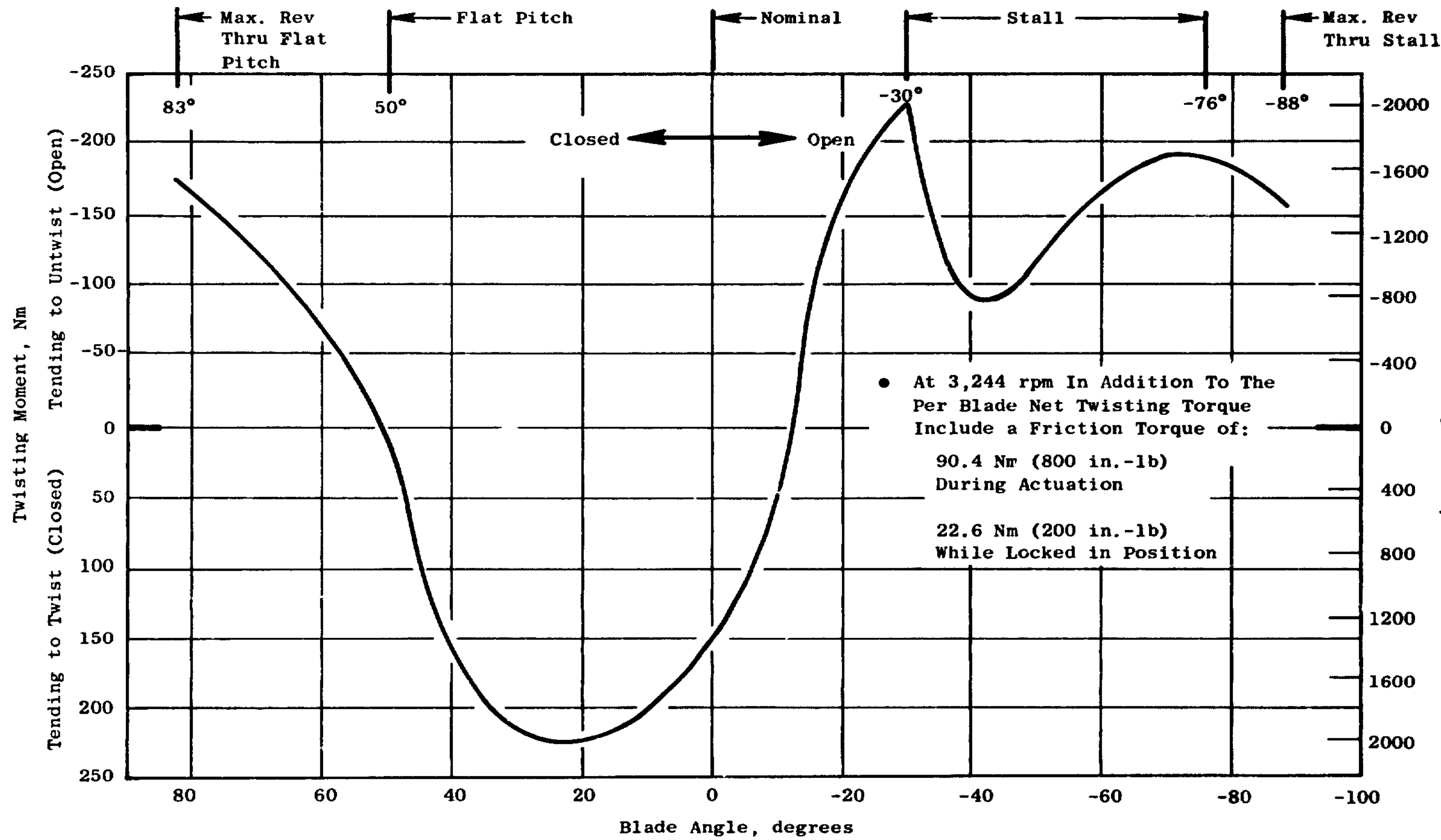


Figure 3. Net per Blade Twisting Torque at 3244 rpm, SLS Standard Day.

4.0 SYSTEM DESCRIPTION

The General Electric ball spline actuation system is shown in cross section in Figure 1, and is illustrated pictorially (except for the drive system) in Figure 4. A functional system schematic also is presented in Figure 5. The actuation system functions in the following manner.

Pinion bevel gears attached to each of the 18 fan blade trunnions are rotated by the motion of two counteracting master ring gears. The pinion bevel gears are attached to the trunnions by accurately positioned fine pitch splines, which allow for proper synchronization with the two ring bevel gears. The fine pitch splines also permit reindexing of the blades to vary the open and closed blade angle limits for engine thrust reversal through both stall and flat pitch. Use of two ring gears permits load sharing, and also adds redundancy to the system. An axial tie member in the area of the pinions prevents separation of the two ring gears. Overall gear ratio of the pinion bevel gear/ring gear mesh is designed to achieve the maximum gear capacity within the space available between any two blades. A shim is provided to ensure proper tooth meshing.

The ring gears are rotated by a ball spline driven by a rigid translating sleeve. The forward ring gear is driven by the outer (straight) portion of the ball spline while the aft ring gear is driven by the inner (helical) portion of the ball spline. As shown in Figure 1, both ring gears are easily removed at their bolted flange joints for modular assembly and disassembly of the actuator.

The rigid translating sleeve of the ball spline is driven by a ball screw through a stroke of 15.49 cm (6.10 in.) to achieve a blade rotation of 135°. The balls of the ball spline ride in a continuous path made up of a loaded track and a return guide. The return guides are tubes located out of the load zone. Loaded tracks and return guides are connected by end return caps. These end caps permit easy replacement of the balls during servicing.

Helical ball tracks between the inner and middle members of the ball spline generate a maximum axial load during normal operation of approximately 108.6 kN (24,398 lb). As illustrated in Figure 6, these axial loads are reacted from the middle member into the ball screw through a ball nut. Ball screw thrust loads are transmitted back into the inner member through a set of precision ground M50 duplex thrust ball bearings. Thus, all high actuator axial loads are close looped on a small diameter within the actuator. This close coupling of high axial loads was instrumental in achieving a low weight for a flight system.

As shown in Figure 1 and illustrated in Figure 5, power to drive the ball screw is provided by a hydraulic motor, acting through a gear differential. A ball/ramp-type no-back is included between the differential gear and the ball screw to allow torque to be transmitted only in one direction. Axial stops at each end of the ball screw (Figure 1) limit actuator travel.

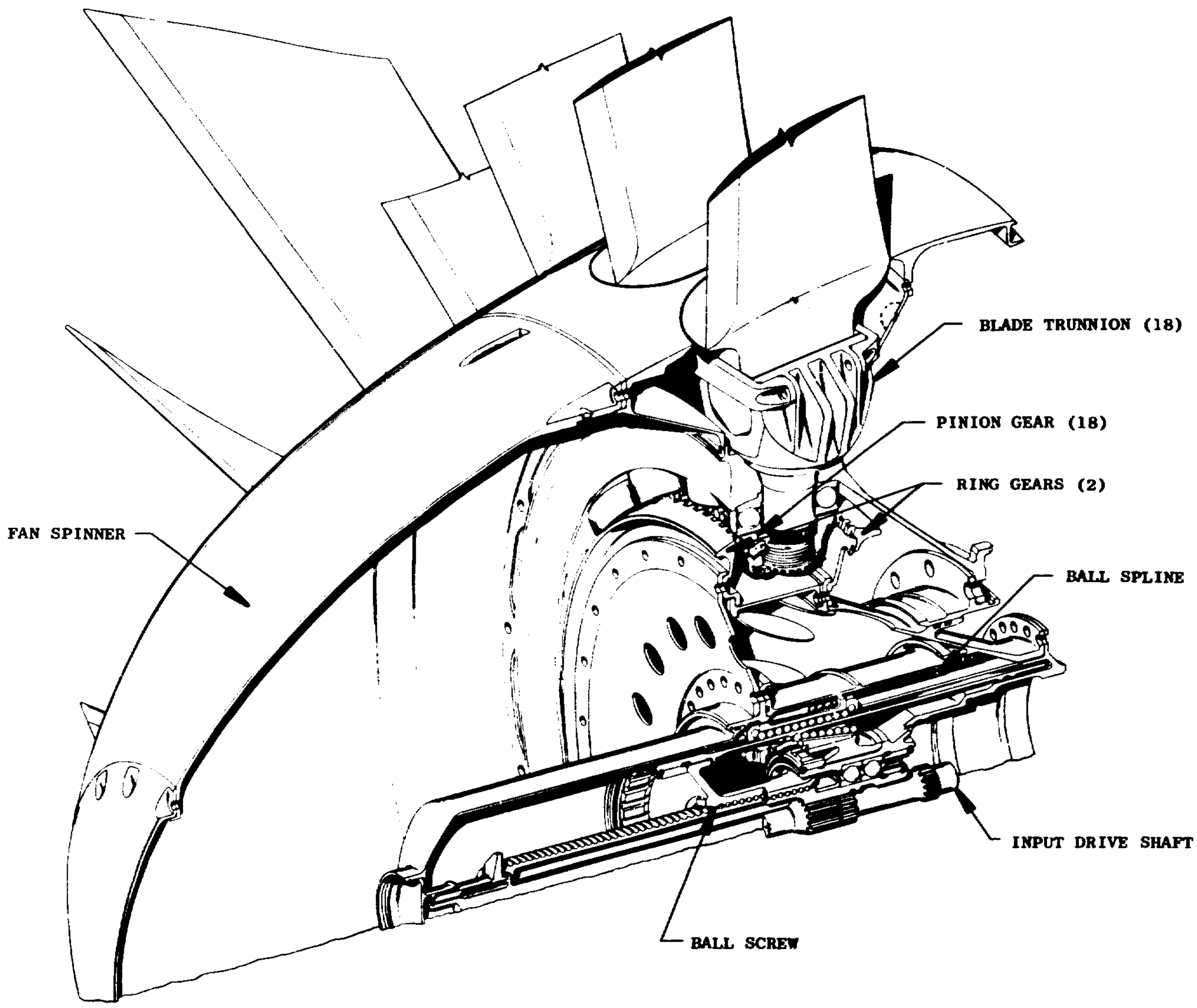


Figure 4. GE Ball Spline Actuator System, (Trimetric).

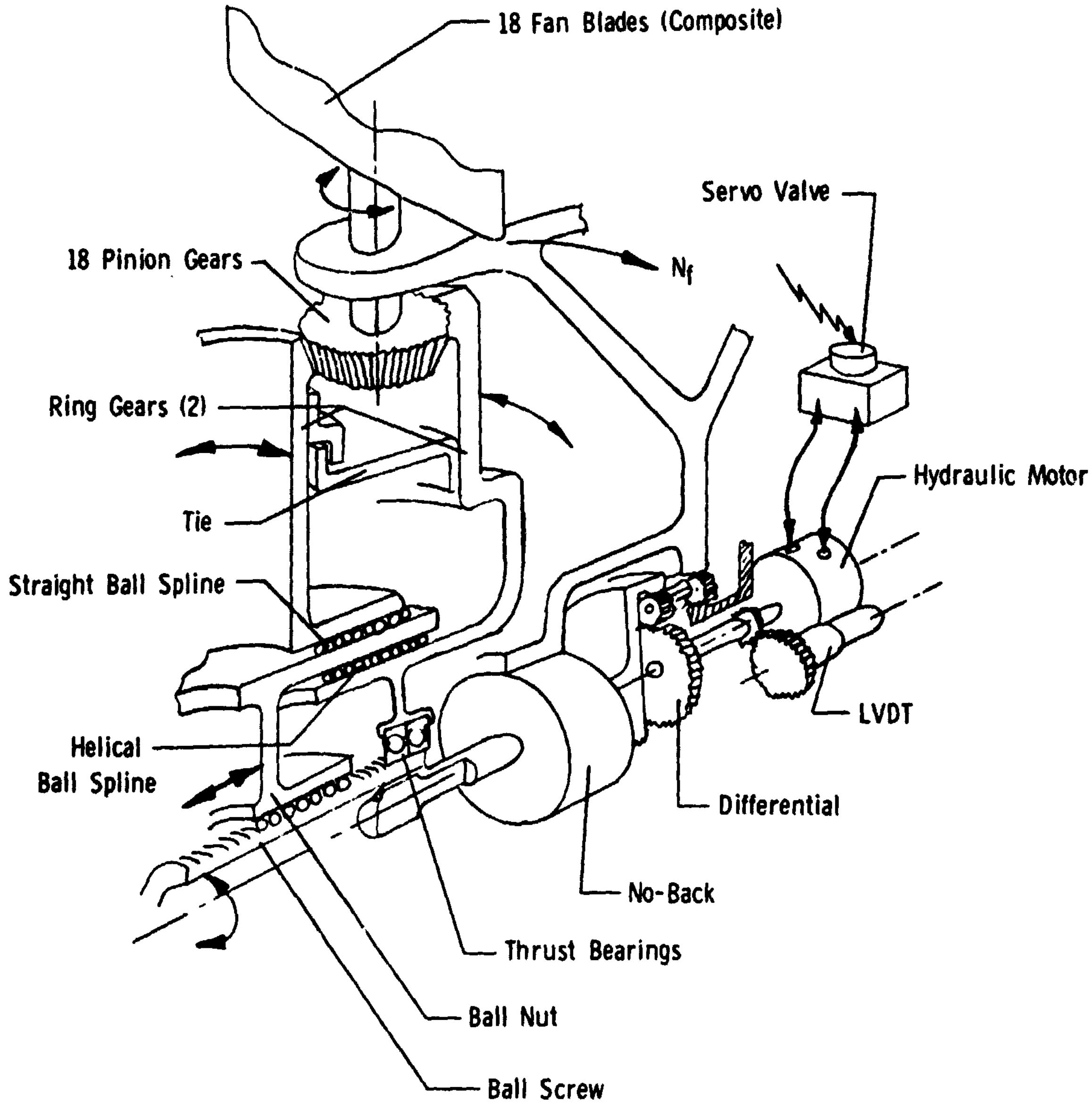
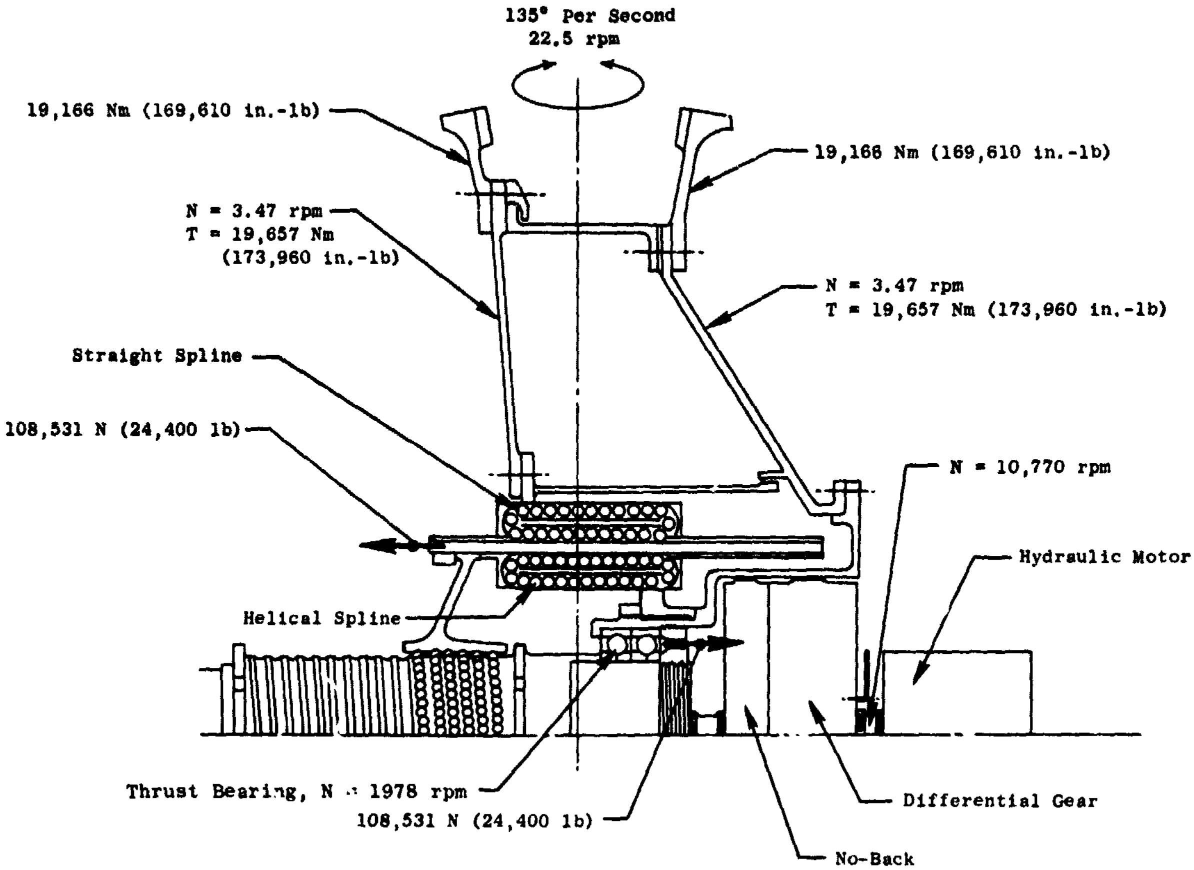


Figure 5. GE Ball Spline Variable-Pitch Mechanism Schematic.



- Per Blade Torque = 318.8 Nm (2821 in.-lb)
- Total Torque At Blades = 5738 Nm (50,778 in.-lb)
- System Efficiency = 66.1%
- System Gear Ratio = 478.68
- Torque At Motor = 18.1 Nm (160.5 in.-lb)

Figure 6. Actuator Design Speed and Load Characteristics.

A summary of the actuator design speed (rpm) and load characteristics is presented in Figure 6. Estimated component efficiency levels utilized to establish the motor power requirements and torque distribution through the system are presented in Table II. The values shown are considered "minimum" levels and were conservatively assumed to establish power requirements and torque levels for steady-state stress analysis. For bird strike calculations, where torque input to the actuation system is from the blade, a higher level of efficiency was assumed for the bevel/pinion gear mesh to establish the maximum possible component loads. The overall system efficiency based on the levels shown in Table II is 66.1%. Also shown in Table II are the gear ratios of individual components. The overall gear ratio is 478.68

Table II. Component Efficiencies.

	Percent	Gear Ratio
Differential Gear	97	5.444
No-Back	98	--
Ball Screw Thrust Bearing	94	--
Ball Screw	92	569.77
Ball Spline	85	
Bevel Gear Thrust Bearing	97.5	--
Bevel/Pinnion Gear Mesh	97	6.480

5.0 COMPONENT DESIGN

5.1 BEVEL GEARS

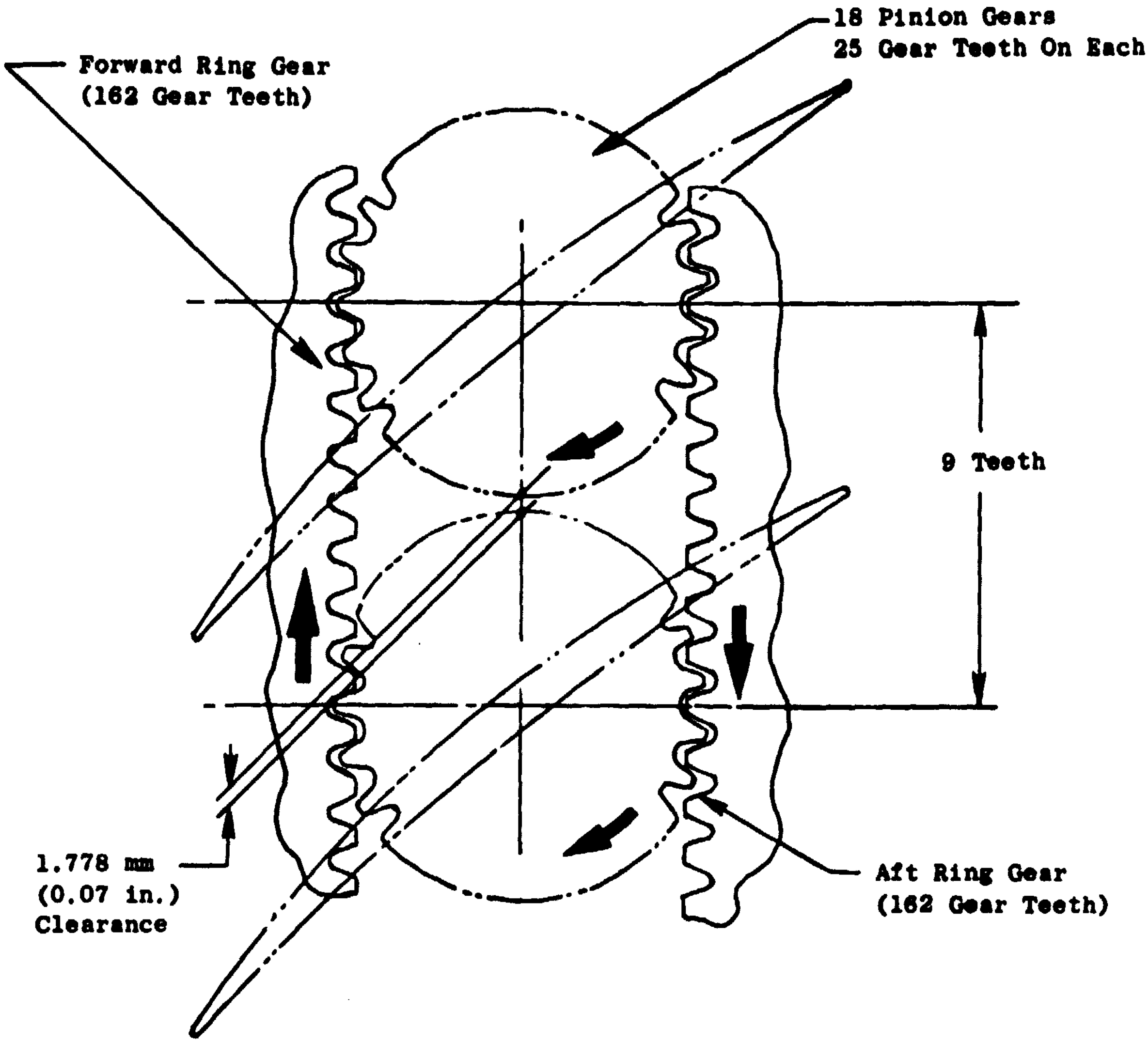
As shown in Figure 1, torque generated by the ball spline is transferred radially outward through cones to the forward and aft ring gears. These ring bevel gears synchronize and drive the 18-blade pinion bevel gears. Transmitting torque from the ball spline through two gear meshes into each pinion gear minimizes the actuator requirements and provides redundancy in the event of excessive blade impact twisting loads.

Geometry of the mating pinion and ring gear teeth is defined in Figure 7. Pinion gear tooth design loads and stress data are presented in Table III. Allowable stress for the condition of maximum operating torque is based on GE design practice for maximum continuous operating bending and compressive stresses. For the bird strike condition (FOD impact load), the allowable bending stress is based on GE design practice for impact loading of gear teeth. Gear tooth stresses in the ring gears would be essentially the same as those shown for the pinion gears.

The ring gear stresses, as shown in Figures 8 and 9, are representative of the most severe loading conditions only; for a typical nominal operating condition the stresses are lower; therefore, no fatigue problems are incurred with this gear design. All stress numbers presented are effective stresses; calculated using the von Mises-Hencky failure criterion and compared directly with the yield strength to obtain margins of safety.

Effective load sharing between the two ring gears requires that the pinions be accurately positioned with reference to the axis of the pinions and the axis of the disk. A study was made to determine the eccentricity of pinion gear true positioning along these axes. The maximum eccentricity along the axes of the disk and pinions are ±0.0035 cm (±0.0014 in.) and ±0.0022 cm (±0.0009 in.), respectively, which are within the allowable limits for hardened and ground gear teeth.

Deflections of the ring gears during normal system operation for conditions of centrifugal and maximum torque loading are shown in Figures 10 and 11 for the forward and aft ring gears, respectively. The ring gears have been designed to match deflections with the pinion gears to ensure proper gear tooth patterns. Using the ring gear radial, axial, and angular deflections from Figures 10 and 11, the relative deflections of the pinion and ring gears were established and are presented in Figure 12. The radial growth and rotational difference between the pinion and ring gears are 0.0022 cm (0.00088 in.) and 0.00005 radians (0.0027°), respectively, for a maximum torque of ±19,199 Nm (±169,610 in.-lb), and a fan rotational speed of 3200 rpm. The gears are to be assembled with zero backlash, and the effects of maximum torque loading and 3200 rpm fan rotation is to increase the gear backlash to 0.0150 cm (0.0059 in.); the backlash of the forward ring gear is within 0.0015 cm (0.00059 in.) of the aft ring gear.



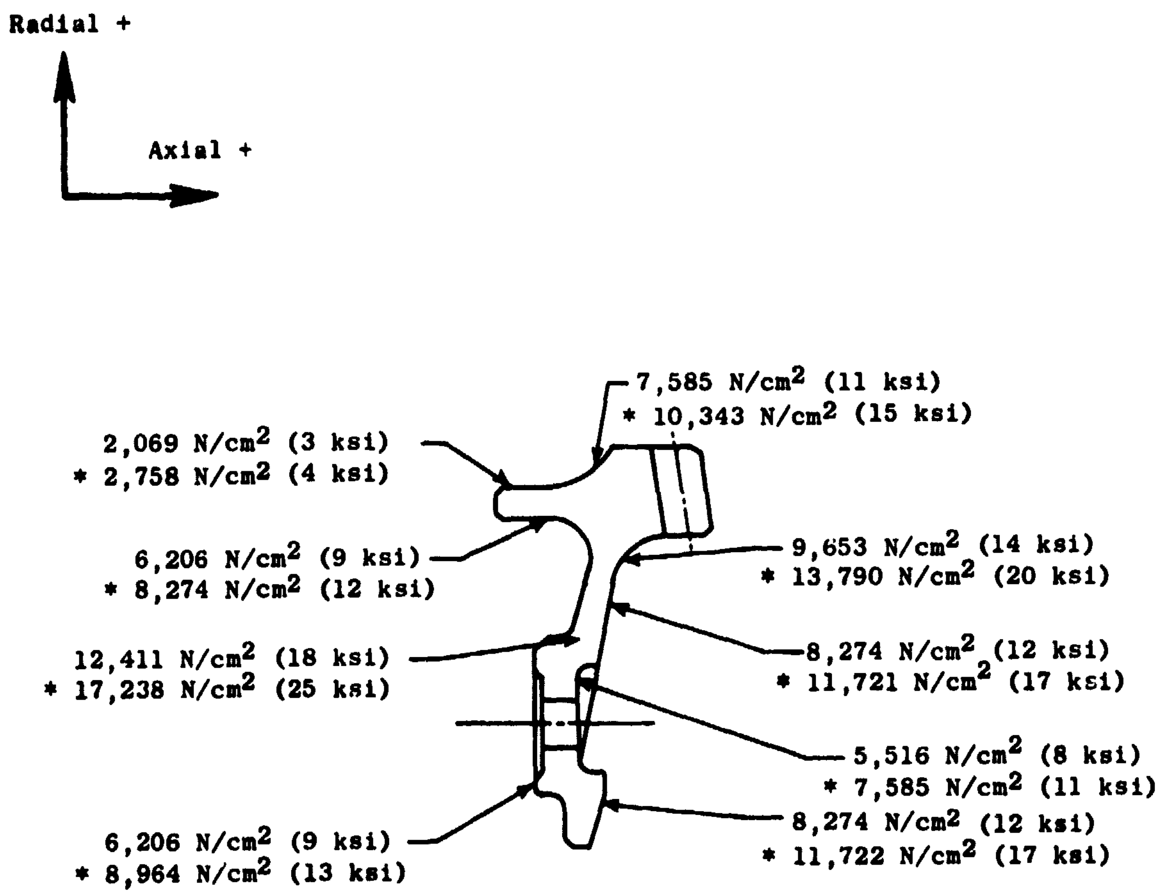
- Gear Ratio = 6.4800
- Diameter Pitch = 8.5443
- Face Width = 1.143 cm (0.450 in.)
- Pressure Angle = 20°
- Tooth Type = Zerol
- Material = AISI 9310

Figure 7. Bevel Gear Geometry.

Table III. Pinion Gear Tooth Design Loads and Stresses.

	Maximum Operating Loads and Stresses (Per Mesh)		Allowable Stresses	
	Condition of Maximum Operating Torque	Bird Strike Condition*	Condition of Maximum Operating Torque	Bird Strike Condition*
Design Loads	15,933 cm-N (1412 in.-lb)	84,750 cm-N (7500 in.-lb)	---	---
Bending Stress	23,443 N/cm^2 (34,000 psi)	99,978 N/cm^2 (145,000 psi)	33,786 N/cm^2 (49,000 psi)	103,425 N/cm^2 (150,000 psi)
Compressive Stress	157,206 N/cm^2 (228,000 psi)	---	172,375 N/cm^2 (250,000 psi)	---

*Bird strike condition is assumed to be an equivalent torque on two adjacent blades of 169,500 cm-N (15,000 in.-lb) each.

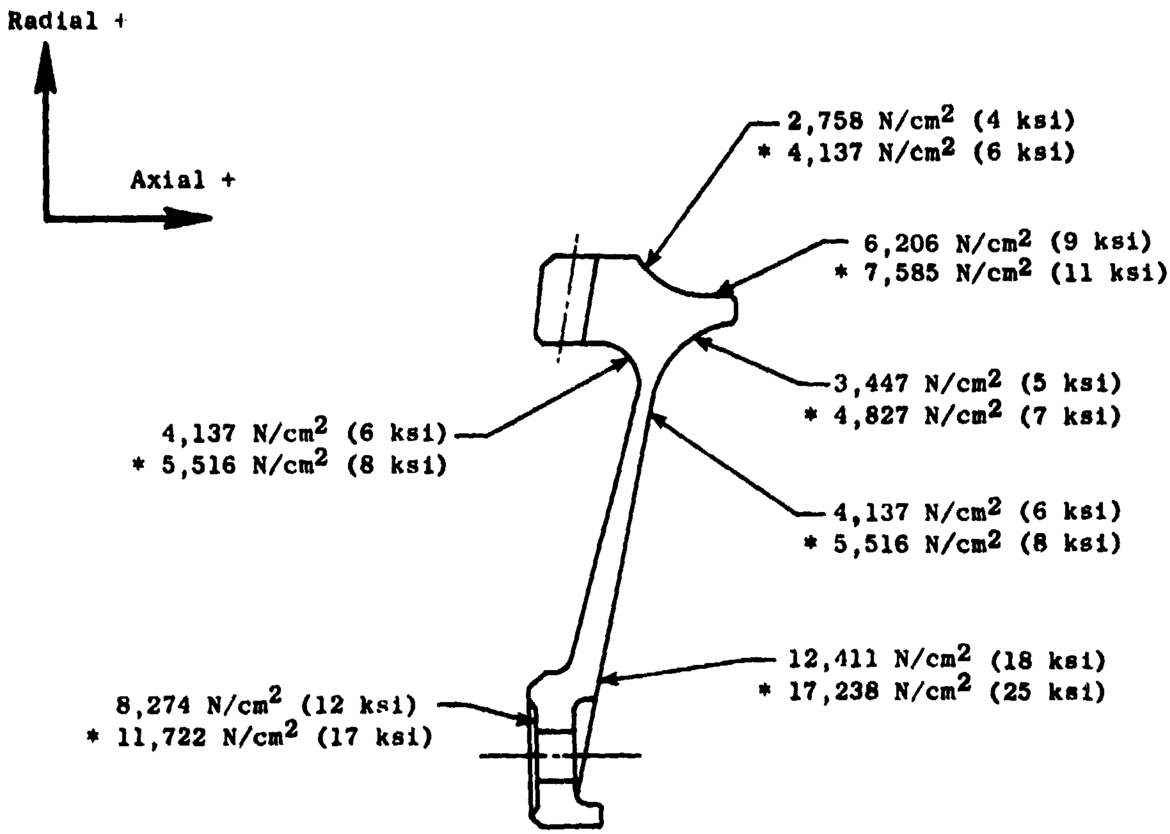


- Loading:

	Maximum Torque	* Bird Strike (FOD Impact Load)
Torque	± 19,166 Nm (±169,610 in.-lb)	±27,509 Nm (±243,440 in.-lb)
Axial	- 29,668 N (- 6,670 lb)	- 42,585 N (- 9,574 lb)
Radial	4,893 N (1,100 lb)	7019 N (1,578 lb)
Speed	3,200 rpm	3,200 rpm

- Material = AISI 9310, 0.2 Yield Strength, S_Y = 95,151 N/cm^2 (138 ksi)

Figure 8. Forward Ring Gear Design Loads and Stresses.

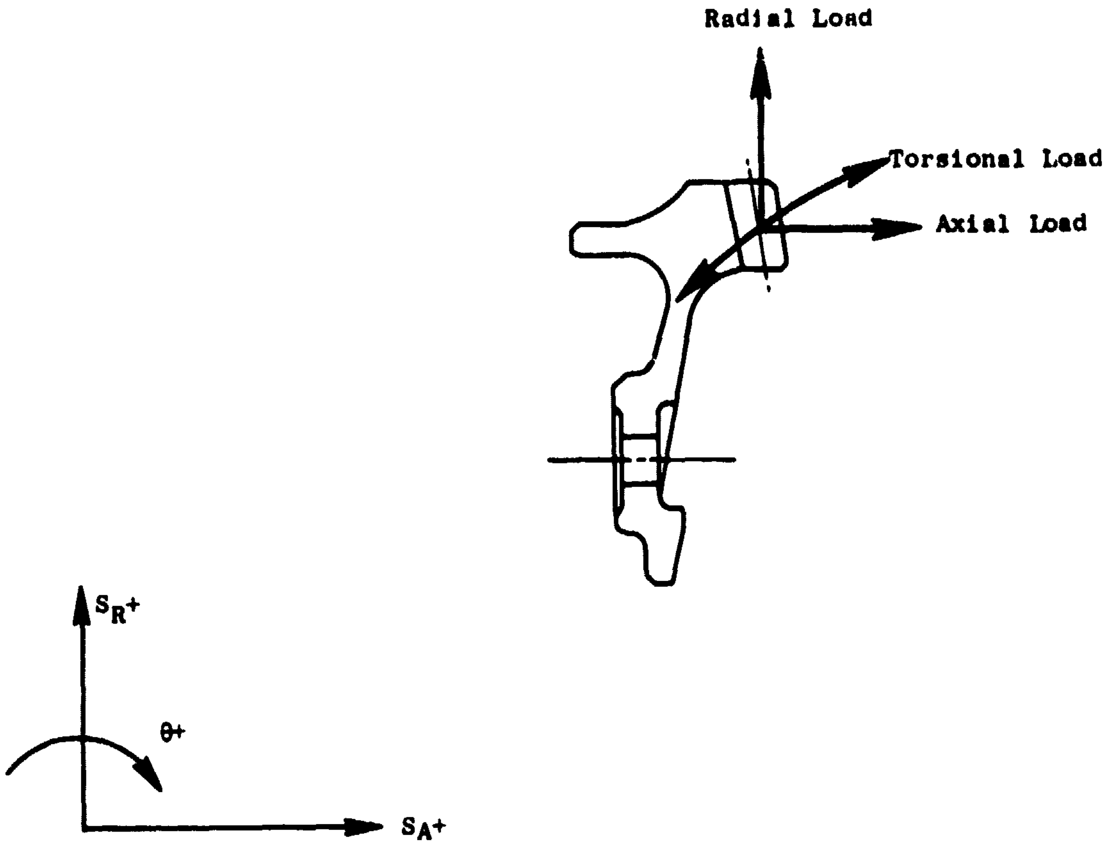


- Loading:

	Maximum Torque	* Bird Strike (FOD Impact Load)
Torque	± 19,166 Nm (±169,610 in.-lb)	± 27,509 Nm (±243,440 in.-lb)
Axial	29,668 N (6,670 lb)	42,585 N (9,574 lb)
Radial	4,893 N (1,100 lb)	7,019 N (1,578 lb)
Speed	3,200 rpm	3,200 rpm

- Material = AISI 9310, Sy = 95,151 N/cm² (138 ksi)

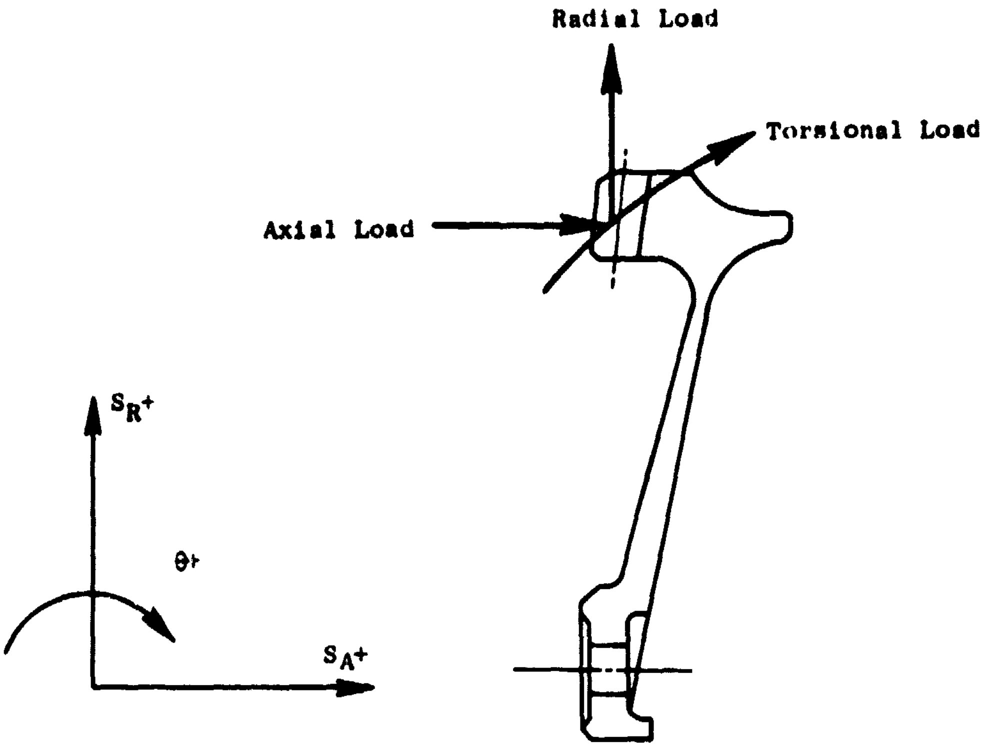
Figure 9. Aft Ring Gear Design Loads and Stresses.



Gear Mesh Deflection

	Centrifugal	Maximum Torque
S_R	0.00518 cm (0.00204 in.)	0.0142 cm (0.00558 in.)
S_A	-0.00340 cm (-0.00134 in.)	-0.0251 cm (-0.0099 in.)
θ	-0.00060 Radian	-0.0067 Radian

Figure 10. Forward Ring Gear Deflections.

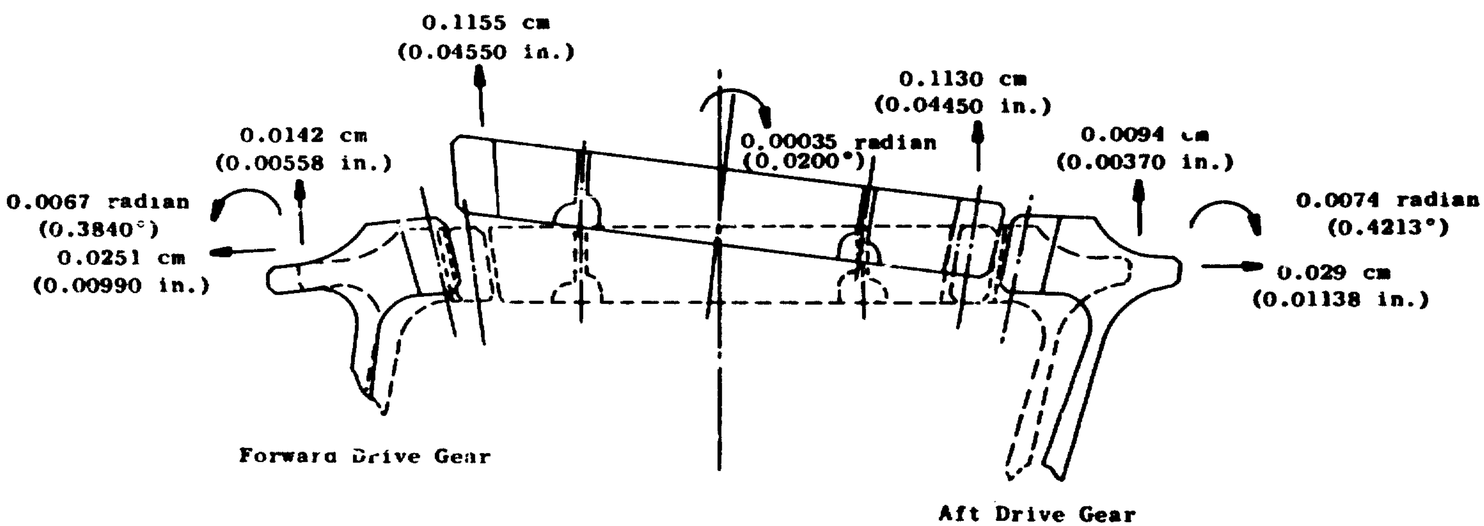


Gear Mesh Deflections

	Centrifugal	Maximum Torque
S_R	0.00376 cm (0.00148 in.)	0.0094 cm (0.0037 in.)
S_A	-0.00551 cm (-0.00217 in.)	0.0290 cm (0.0114 in.)
θ	-0.00057 Radian	0.0074 Radian

Figure 11. Aft Ring Gear Deflections.

PINION MOVEMENT



DEFLECTIONS

MAXIMUM TORQUE CONDITION

±19,166 Nm (±169,610 in.-lb) @ 3200 rpm

DIFFERENCES IN RADIAL GROWTH OF PINION AND GEARS

Forward Gear	Aft Gear
0.1014 cm (0.03992 in.)	0.1036 cm (0.0408 in.)

Δ = 0.0022 cm (0.00088 in.)

DIFFERENCES IN ROTATION OF PINION AND GEARS

Forward Gear	Aft Gear
0.00705 radian (0.4040°)	0.007 radian (0.4013°)

Δ = 0.00005 radian (0.0027°)

Figure 12. Pinion Movement, Max. Torque Loads.

Under torsional loading conditions (i.e., when the actuation system is being utilized to either maintain or change the blade pitch angle), the actuator will remain centered about its own axis. However, when the fan blades pass through zero net twisting torque, gear tooth loads are not being applied at the ring gear/pinion mesh and the actuation system center of gravity can shift off-center due to increased gear tooth backlash which results from centrifugal loading. This condition has been evaluated using data shown in Figure 13 and its possible effect on system unbalance has been determined. The maximum center-of-gravity displacement that can occur, based on the increased gear tooth backlash, is 0.003 cm (0.0012 in.). This displacement is well within system unbalance limits.

5.2 AXIAL SUPPORT LINK

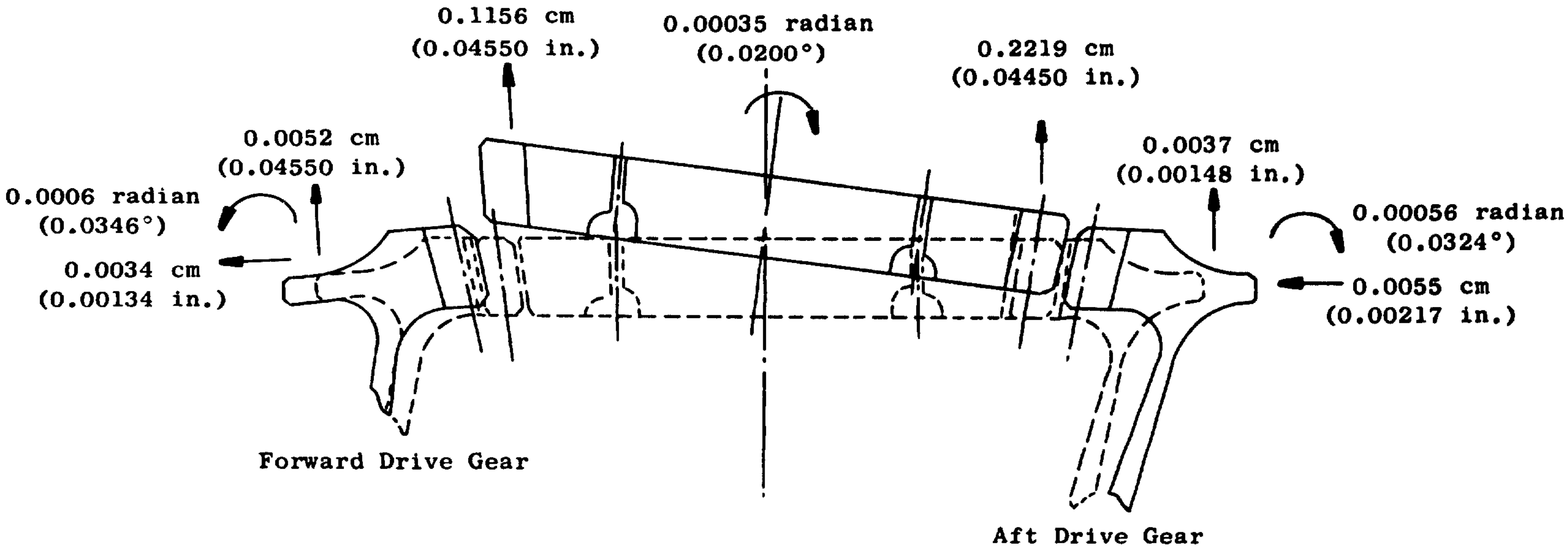
As shown in Figure 1, the axial support link ties the forward and aft ring gears together providing a close-coupled load path for reacting the ring gear separating loads. Resultant effective stresses, calculated using the von Mises-Hencky failure criterion, are presented in Figure 14 for conditions of maximum gear torque and bird strike. The residual torque shown for the load conditions is a result of the ring gear thrust bearing efficiency discussed below.

5.3 GEAR SLIDING THRUST BEARING

Since there is relative motion between the forward ring gear and the aft ring gear, a sliding thrust bearing is provided at the forward ring gear and axial support link interface. Bearing design loads equivalent to gear separating loads for maximum operating torque and bird strike are presented in Figure 15. Also shown in Figure 15 is the available friction test data for Fibriloid. The test data were used for the most severe conditions of low loads and low temperatures to arrive at a coefficient of friction for the bearing of 0.087 which, in conjunction with the maximum design loading, yields a bearing efficiency of 0.975. Also shown in Figure 15 is the calculated maximum bearing axial load of approximately 42,585 N (9574 lb), which can be compared on a load/area basis with the wear test data of 0.00381 cm (0.0015 in.) resulting from 100,000 cycles of load/area equivalent to 27,580 N/cm^2 (40,000 psi) and subjected to a temperature of 436 K (325° F). Consideration also has been given to the use of Karon as a substitute or prime material to be used for this bearing application, since component testing has shown that the Karon yields both coefficients of friction and resistance to wear of equivalent magnitude to that of the Fibriloid.

5.4 RING GEAR DRIVE CONES

Torque is transmitted to the ring gears from the ball spline through forward and aft drive cones attached to the outer and inner diameter ball spline members, respectively. The majority of load on these drive cones is



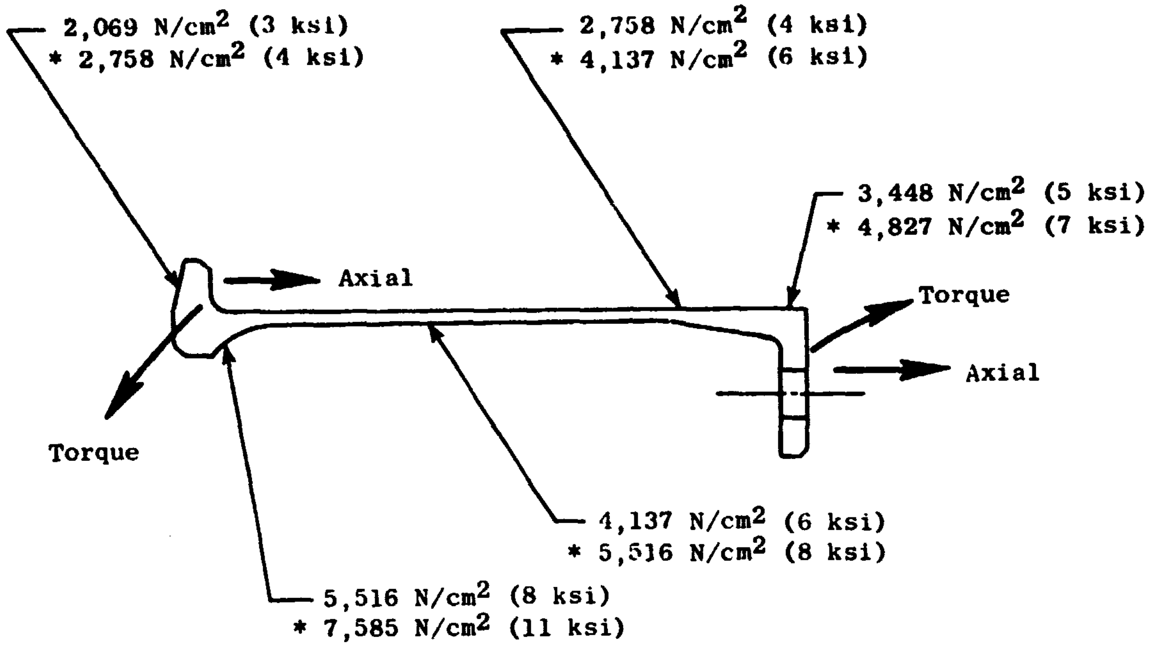
DEFLECTIONS
CENTRIFUGAL LOAD

@ 3200 rpm

DIFFERENCES IN RADIAL GROWTH
OF PINION AND GEARS

Forward Gear	Aft Gear
0.1104 cm (0.04346 in.)	0.1168 cm (0.04598 in.)

Figure 13. Pinion Movement, Centrifugal Load.



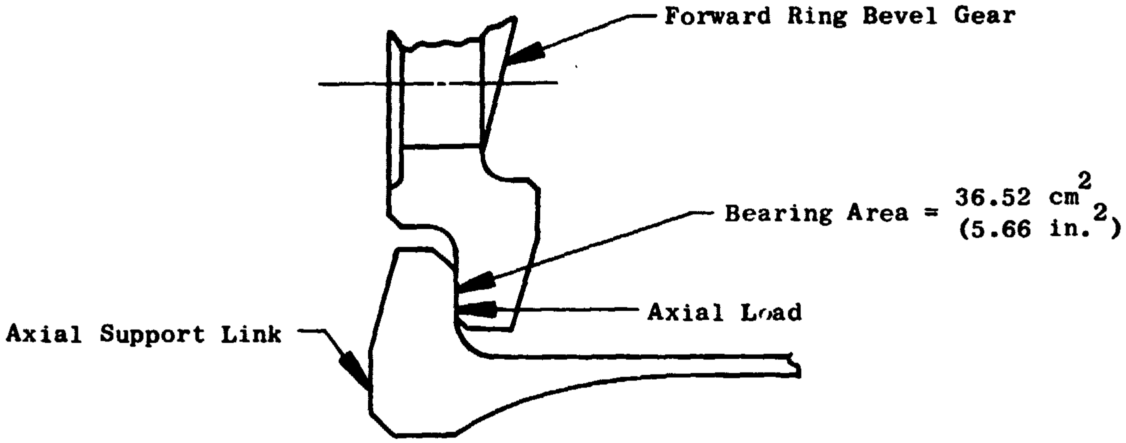
- Loading

	Maximum Torque	* Bird Strike (FOD Impact Load)
Torque	491 Nm (4,349 in.-lb)	705 Nm (6,242 in.-lb)
Axial	29,668 N (6,670 lb)	42,585 N (9,574 lb)
Speed	3200 rpm	3200 rpm

- Material = Ti 6-4, 0.02 Yield Strength, S_Y= 66,881 N/cm^2 (97 ksi) at 328 K (130° F)

Note: Assumed Torque from Loss in Bearing (2.5%)

Figure 14. Axial Support Link Design Loads and Stresses.



- **Loading:**

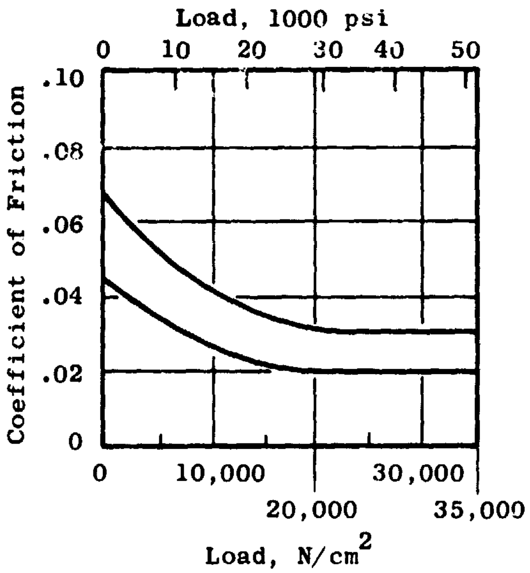
	Maximum Torque	Bird Strike (FOD Impact)
Axial Load	29,668 N (6670 lb)	42,585 N (9574 lb)
Load/Area	829 N/cm^2 (1202 psi)	1189 N/cm^2 (1725 psi)

- **Material = Fibriloid or Karon**
- **Test Data:**

  Load/Area = 27,580 N/cm^2 (40,000 psi) at 436 K (325° F)
  Wear = 0.0381 mm (0.0015 inch) for 100,000 Cycles

FIBRILOID

Friction Versus Load



FIBRILOID

Friction Versus Temperature

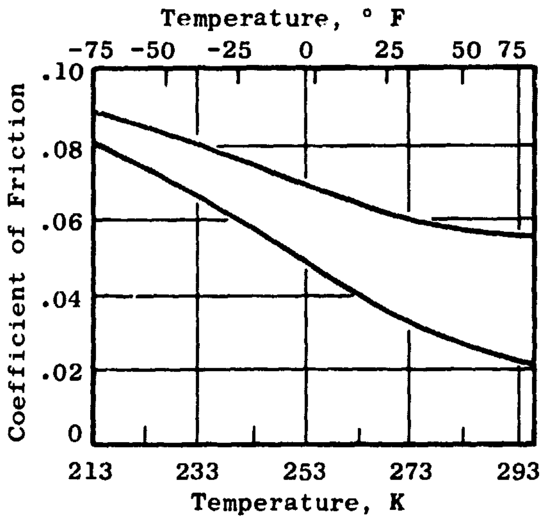


Figure 15. Ring Gear Sliding Thrust Bearing Design Loads.

the torque developed at the ring gears and the torque required to overcome friction in the ring gear sliding thrust bearing. The loads and resultant stress (von Mises-Hencky effective stresses) for the conditions of maximum operating torque and bird strike (FOD impact loading) are presented in Figures 16 and 17. As shown, overall stress levels are low with the exception of the area around the access holes, these holes being required for modular removal of actuation and fan rotor disk. In these regions, high stress levels are encountered due to the combination of large meridional, hoop, and torsional stress concentration factors. It is to be noted that the stresses are compared to 0.02% yield strength values and not 0.2%; the higher yield strength (0.2%) would produce 15% higher margins of safety.

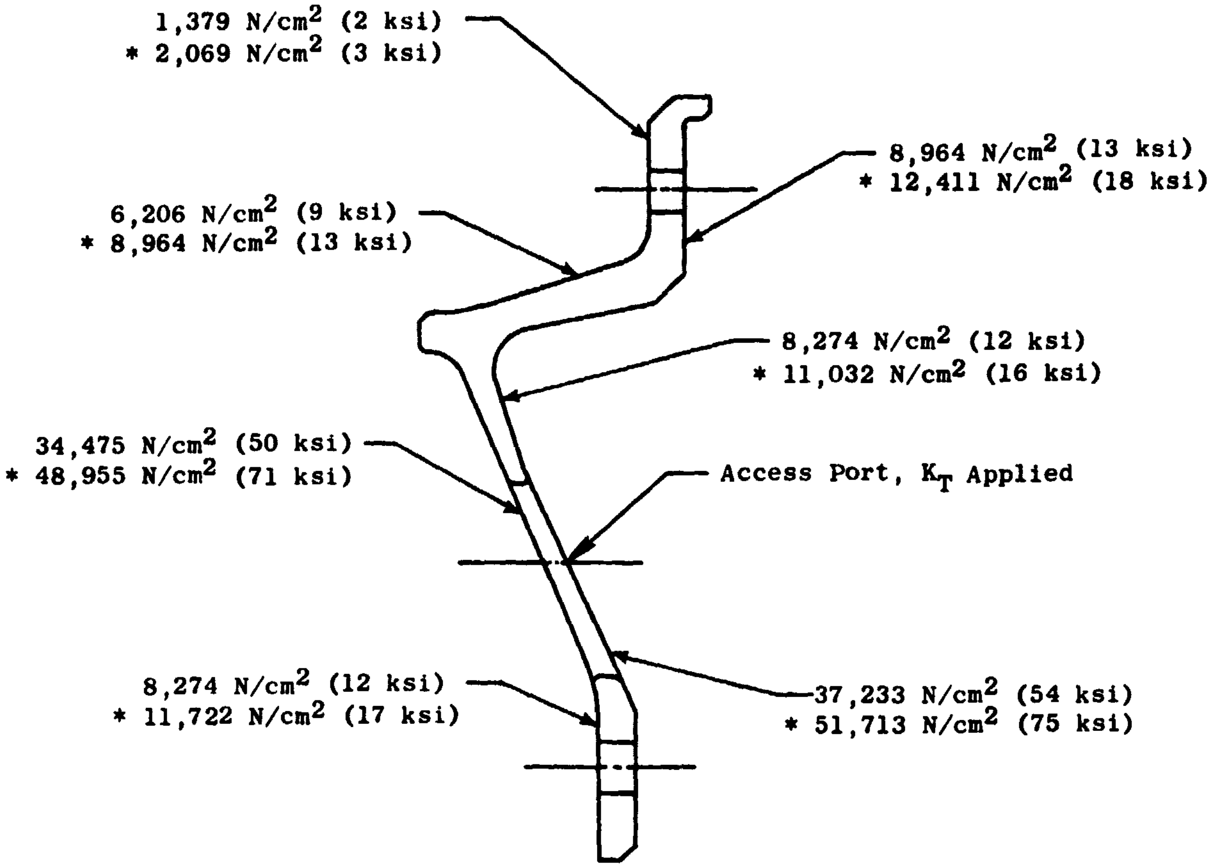
5.5 BALL SPLINE

The ball spline included in the UTW actuation system is a scaled version of a proven design used in two previous General Electric variable-pitch fan blade test vehicles. As shown in Figure 18, the ball spline is basically a three-piece configuration, consisting of the outer and inner diameter members whose motion is tangentially opposite, and the fore and aft translating midmember. There are 12 recirculating linear ball tracks between the outer and middle member and 12 recirculating helical tracks between the middle and inner members. Each of the circuits contains 144 active balls, 0.635 cm (0.25 in.) in diameter.

The maximum normal operating design torque of the ball spline is 19,659 m-N (174,000 in.-lb). Maximum steady-state capacity is 23,391 m-N (207,000 in.-lb), providing a 19% design margin. Under bird strike (FOD impact) conditions, the ball spline is subjected to a restraining torque of 27,459 m-N (243,400 in.-lb) with a maximum impact load capacity of 39,810 m-N (352,300 in.-lb), providing a 45% design margin.

To meet 36,000 hours of engine life, a total linear travel of 2.581×10^6 cm (1.016×10^6 in.) based on the mission duty cycle shown in Figure 2 is required. This includes an allowance of 180° of blade angle travel for modulation during approach for each flight cycle. The calculated design life in total linear travel, based on the cubic mean torque load and the torque capacity of the ball spline, is 1.270×10^6 cm (0.50×10^6 in.). The capacity of the ball spline is determined from design criteria developed by Saginaw Steering Gear, Saginaw, Michigan. This indicates that the balls will only have to be changed once during the life time of the engine. This exceeds the 9000 hours replaceable parts requirement by a factor of two. If the blades are fixed in position during approach, the life of the ball spline exceeds the total 36,000 hour engine life requirement.

Design loads and corresponding stresses for the outer, mid, and inner members of the ball spline are presented in Figures 19, 20, and 21, respectively. As shown in the figures, all stresses are well within the allowable design limits.



- Loading

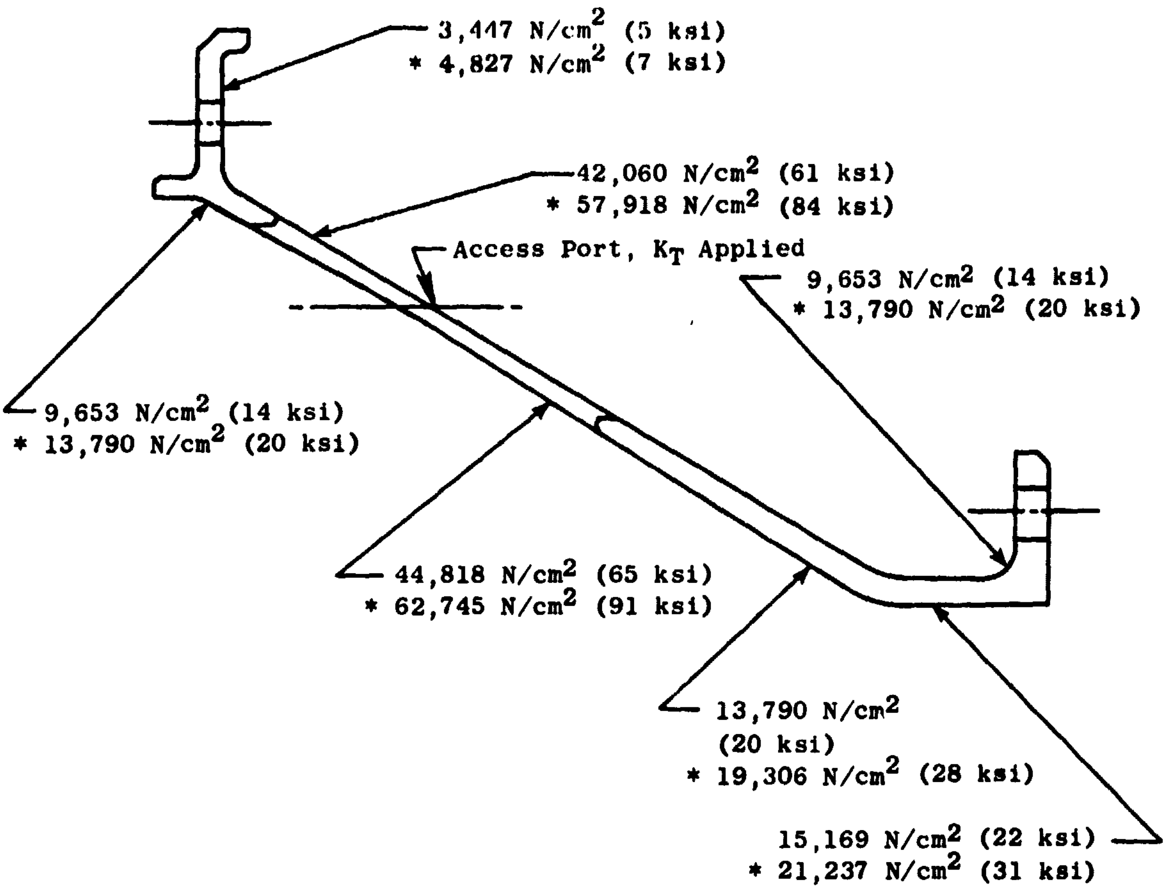
	Maximum Torque	* Bird Strike (FOD Impact Loads)
Torque	19,657 Nm (173,960 in.-lb)	27,509 Nm (243,440 in.-lb)
Speed	3,200 rpm	3,200 rpm

- Material = Ti 6-4, 0.02% Yield Strength, S_Y = 66,881 N/cm^2 (97 ksi) at 328 K (130° F)

Note: Assumed Bird Strike Torque Has No Gear or Bearing Losses.

- $K_{T_{Meridional}} = 3.0$
- $K_{T_{Hoop}} = 3.0$
- $K_{T_{Shear}} = 4.86$

Figure 16. Forward Drive Cone Design Loads and Stresses.



• Loading	Maximum Torque	* Bird Strike (FOD Impact Load)
Torque	19,657 Nm (173,960 in.-lb)	27,509 Nm (243,440 in.-lb)
Speed	3,200 rpm	3,200 rpm

- Material = Ti 6-4, 0.02% Yield Strength, S_Y = 66,881 N/cm^2 (97 ksi) at 328 K (130° F)

Note: Assumed Bird Strike Torque Has No Gear or Bearing Losses.

- $K_{T_{Meridional}} = 3.0$ • $K_{T_{Hoop}} = 3.0$ • $K_{T_{Shear}} = 4.86$

Figure 17. Aft Drive Cone Design Loads and Stresses.

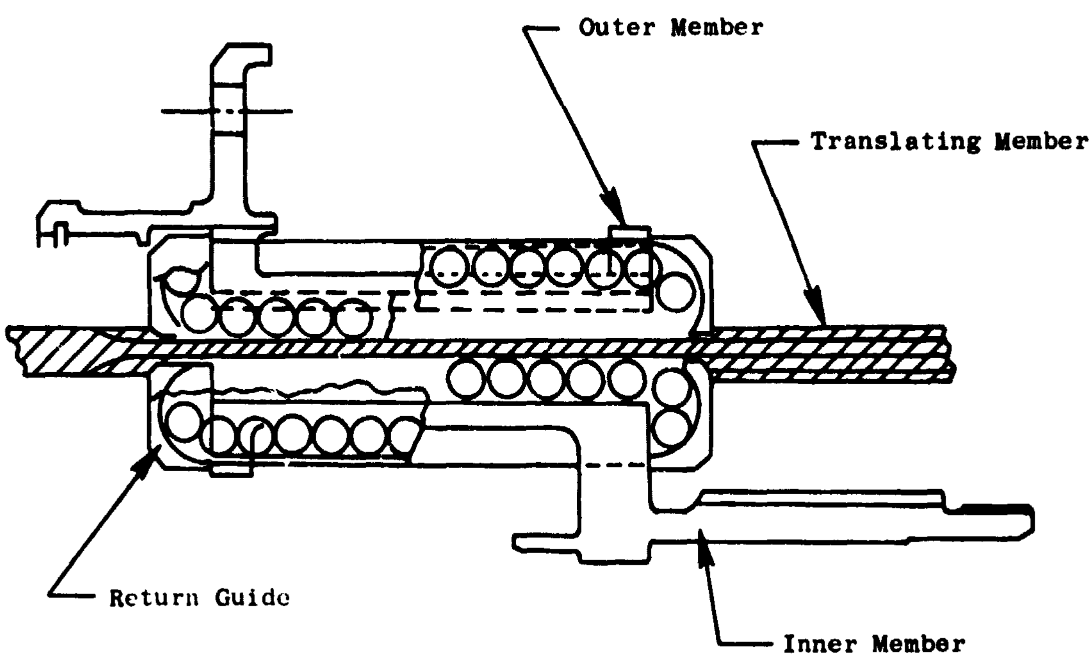
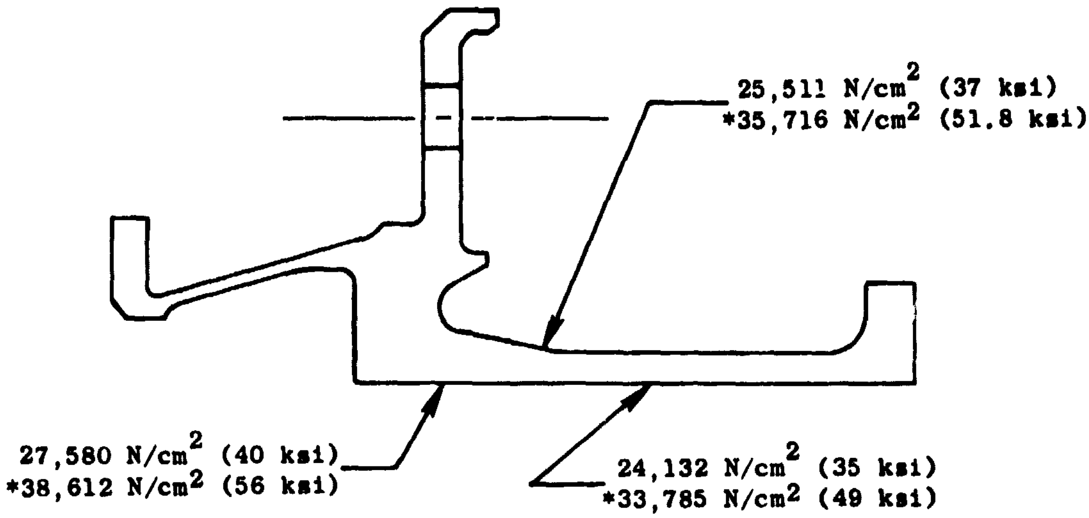


Figure 18. Ball Spline Assembly.



- Loading

  Max. Torque 19,657 Nm (173,960 in.-lb)

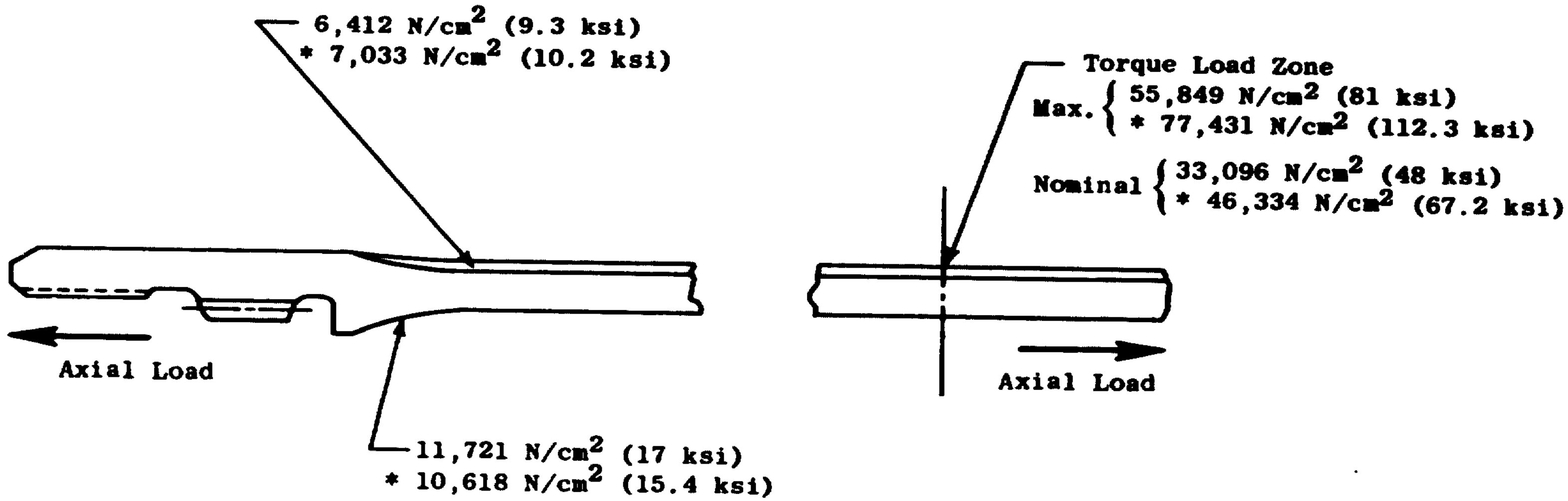
  *Bird Strike 27,509 Nm (243,440 in.-lb)

  Speed 3,200 rpm

- Material

  AISI 9310; S_Y = 95,840 N/cm^2 (139 ksi)

Figure 19. Ball Spline Outer Diameter Member Design Loads and Stresses.



- **Loading**

  Maximum Torque 19,657 Nm (173,960 in.-lb) between Outer Diameter and Inner Diameter Members Only

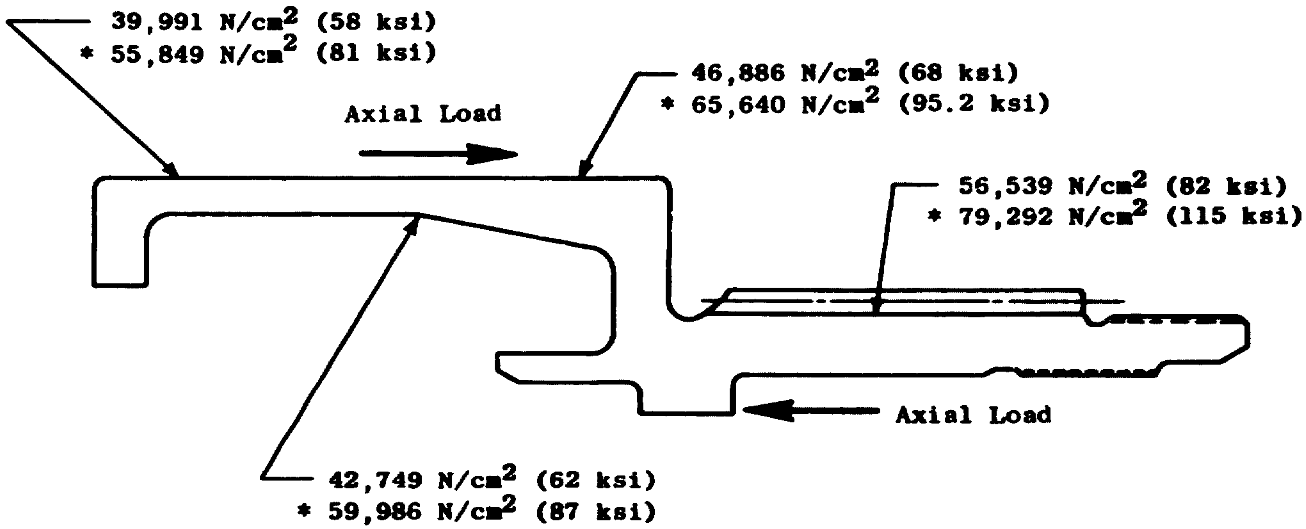
  Maximum Axial Load 108,531 N (24,400 lb)

  * Bird Strike Axial Load 129,090 N (29,022 lb)

  Speed = 3,200 rpm

- Material = AISI 9310, S_Y = 95,840 N/cm^2 (139 ksi)

Figure 20. Ball Spline Midmember Design Loads and Stresses.



- Loading:

	Maximum Torque	* Bird Strike (FOD Impact Load)
Torque	19,657 Nm (173,960 in.-lb)	27,509 Nm (243,440 in.-lb)
Axial	108,531 N (24,400 lb)	129,090 N (29,022 lb)
Speed	3,200 rpm	3,200 rpm

- Material = AISI 9310, S_Y = 95,840 N/cm^2 (139 ksi)

Figure 21. Ball Spline Inner Diameter Member Design Loads and Stresses.

A study was conducted to evaluate the moment distribution in the ball spline midmember, and to select the locations for the oil lubrication holes. Figure 22 is a plan view of the midmember, and shows the straight ball tracks located on the outer cylindrical surface and the helical tracks located on the inner cylindrical surface. The track crossover points form a zero moment section. During the initial ball spline design phase, the ball track configuration was selected so that a zero moment section coincided with the aft end of the ball spline midmember. This was done to minimize the deflection of the aft section, which is considerably more flexible than the forward end.

The oil lubrication holes have been placed where the moment distribution of the outer track superimposed on the moment distribution of the inner track cancel each other. This is shown as the "moment crossover" lines in Figure 22.

5.6 BALL SCREW ASSEMBLY

The ball screw, shown in Figures 1 and 23, is splined to the output shaft of the no-back. Rotation of the ball screw translates the ball nut along the axis of the engine. The ball nut, in turn, provides axial translation of the ball spline middle member.

The ball screw has a 5.08-cm (2.00-in.) pitch diameter with a 0.47-cm (0.185-in.) thread lead, and contains 12 ball circuits having 1.5 turns per circuit.

Previous General Electric variable-pitch fan test vehicles used hydraulic pistons to provide the axial translation required for the ball spline midmember. Application of a ball screw in the UTW system design permitted a reduction in the overall diameter of the ball spline.

To meet the 9000-hour maintainability requirement for bearings and nonreusable parts, based on the mission duty cycle of Figure 2, a total linear travel of 0.645×10^6 cm (0.254×10^6 in.) is required. This includes an allowance of 180° of blade travel for modulation during approach for each flight cycle. The calculated design life in total linear travel based on the cubic mean axial load and the axial load capacity of the ball screw is 0.685×10^6 cm (0.270×10^6 in.) which meets the design requirement. The axial load capacity is based on design criteria developed by Saginaw Steering Gear, Saginaw, Michigan. If the blades are fixed in position during approach, the life of the ball screw exceeds the total 36,000-hour engine life requirement. Under bird strike (FOD impact) conditions, the ball screw is subjected to an estimated axial load of 129,000 N (29,022 lb). The static axial load capacity of the ball screw is 429,517 N (96,564 lb) which is 3.33 times the calculated requirement.

Design loads and resulting stresses for the ball screw and ball nut are presented in Figures 24 and 25, respectively.

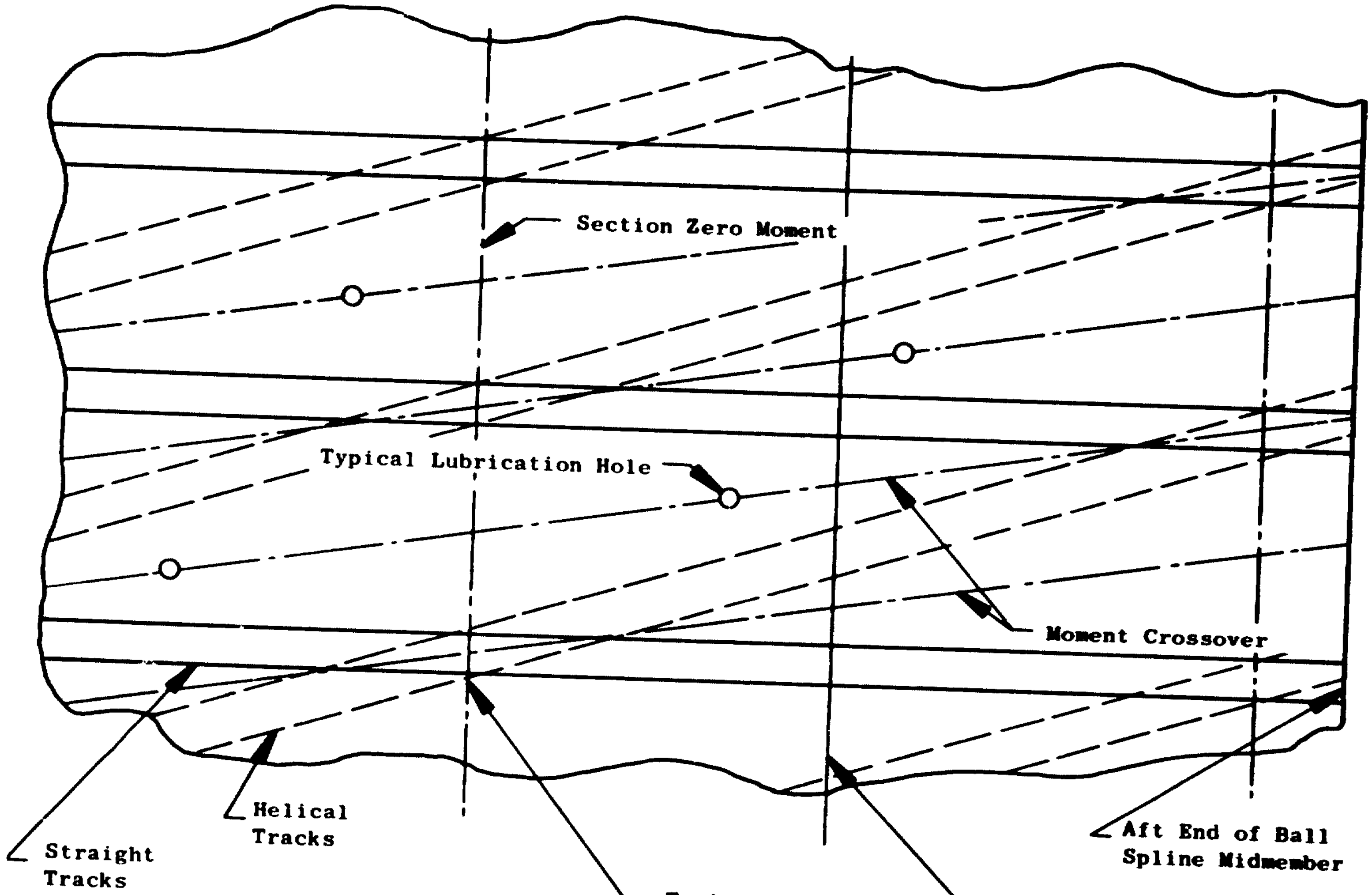


Figure 22. Bal. Spline Midmember Moment Distribution.

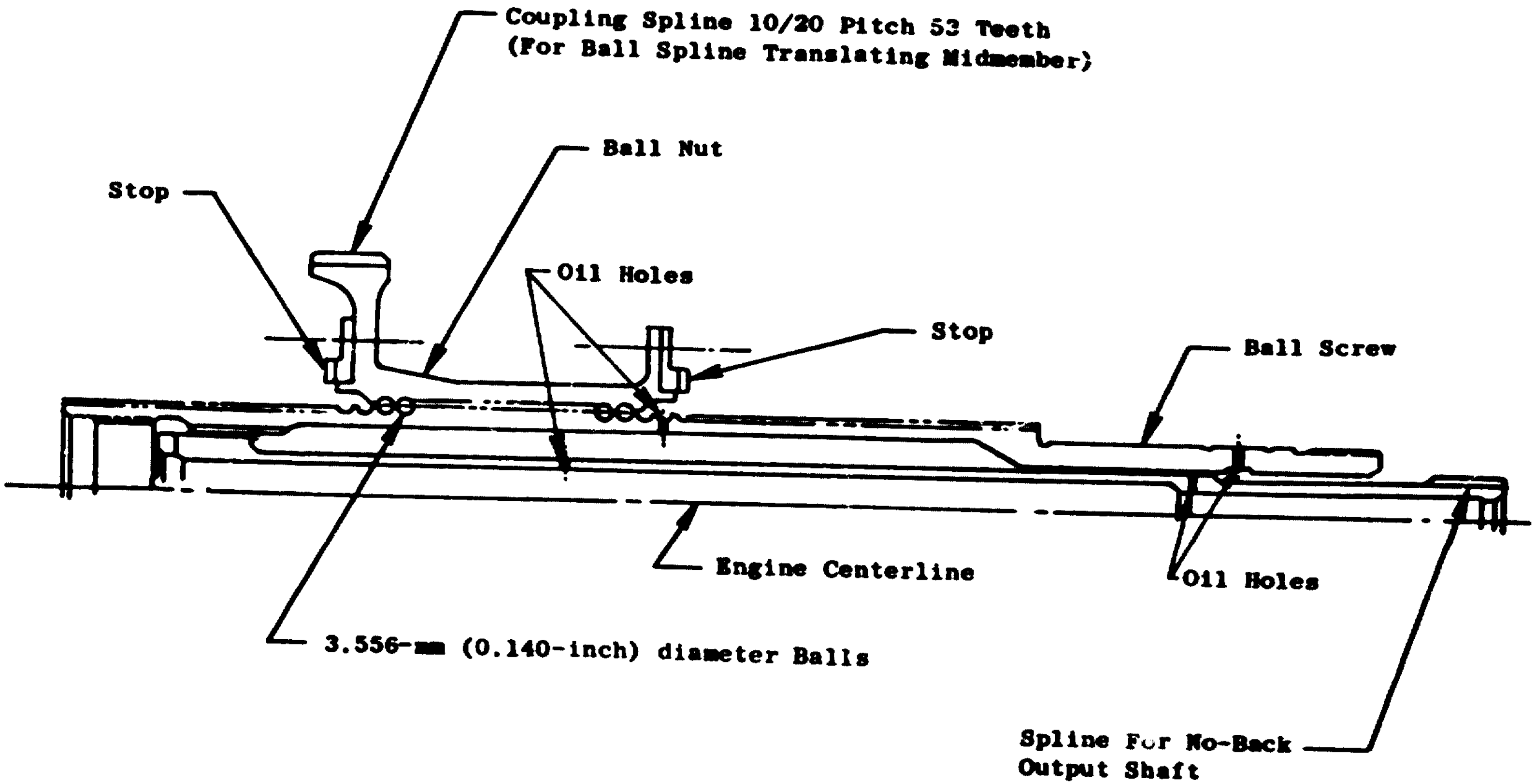
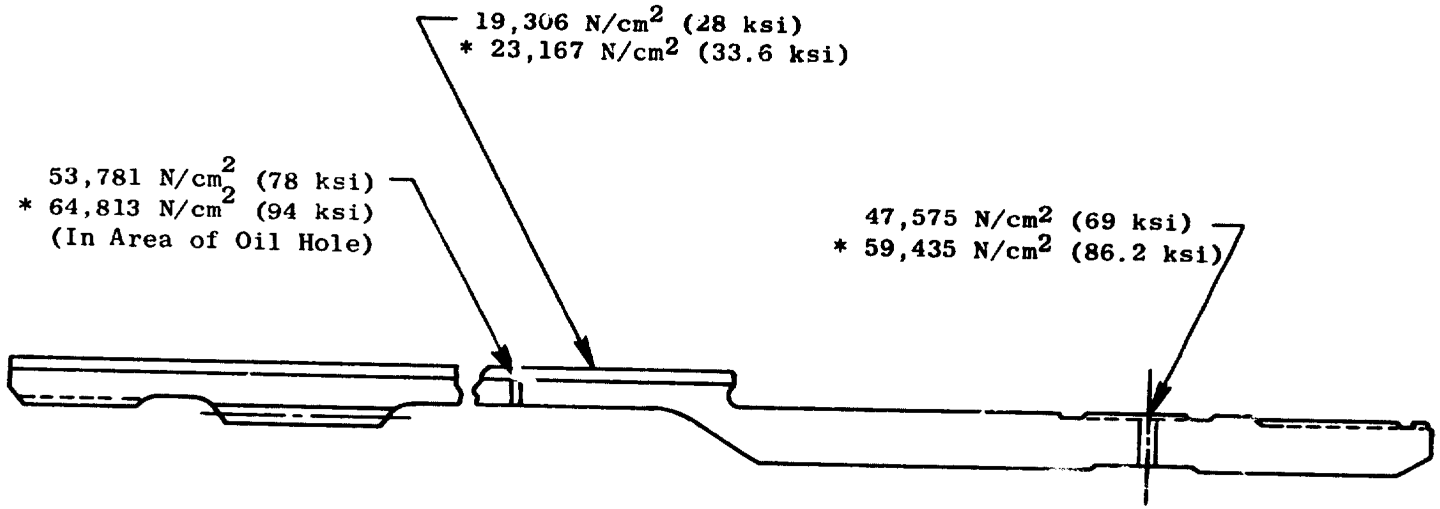


Figure 23. Ball Screw Assembly.

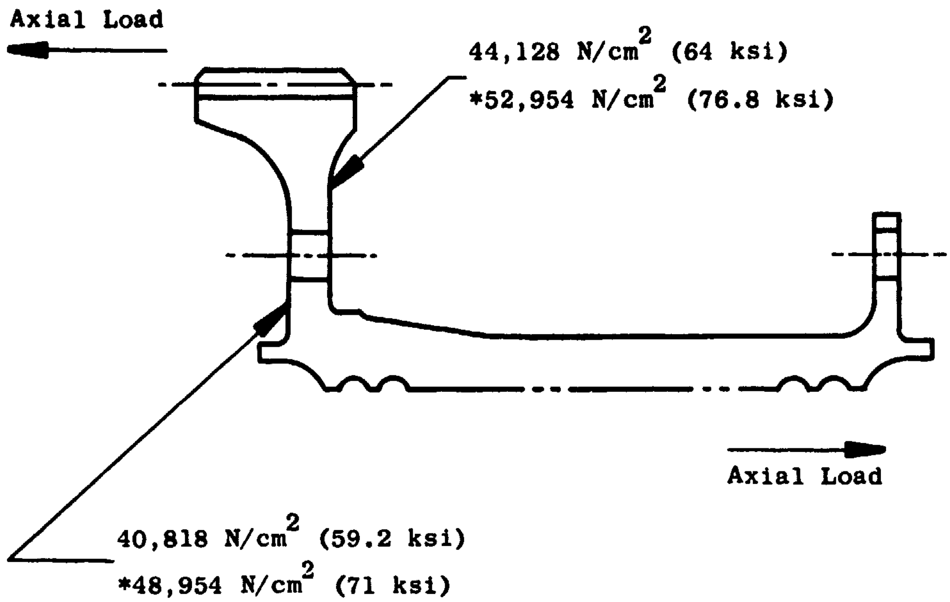


- Loading:

	Maximum Torque	* Bird Strike (FOD Impact Load)
Torque	88.2 Nm (780.9 in.-lb)	96.6 Nm (854.5 in.-lb)
Axial Load	108,531 N (24,400 lb)	129,090 N (29,022 lb)
Speed	3,200 rpm	3,200 rpm

- Material = AISI 9310, S_Y = 95,840 N/cm^2 (139 ksi)

Figure 24. Ball Screw Design Loads and Stresses.



- **Loading**

  Maximum Axial Load 108,531 N (24,400 lb)
  *Bird Strike Axial Load 129,090 N (29,022 lb)
  Speed 3200 rpm

- **Material**

  AISI 9310; S_Y = 95,840 N/cm^2 (139 ksi)

Figure 25. Ball Nut Design Loads and Stresses.

5.7 BALL SCREW THRUST BEARING

The ball screw duplex thrust ball bearings, shown in Figure 1, react the ball nut and ball spline axial load. Design data for the bearing are presented in Figure 26. Bearing races are (assembled and) flush ground (by the bearing vendor) to maximize load-sharing capability. Bidirectional load sharing also is improved by close-matching of the contact angles. Although additional bearing race shoulder height was required to accommodate the high thrust loads, there is still sufficient radial space for an adequate cage cross section.

As shown in Figure 26, the calculated bearing B_{10} life is considerably in excess of the required B_{10} life to satisfy the full 48,000 engine mission cycles.

5.8 DYNAMIC STOPS

As shown in Figure 1, axial dynamic stops are included at each end of the ball screw travel. These stops are designed to absorb the full dynamic load transmitted by the ball nut in the event of a system failure.

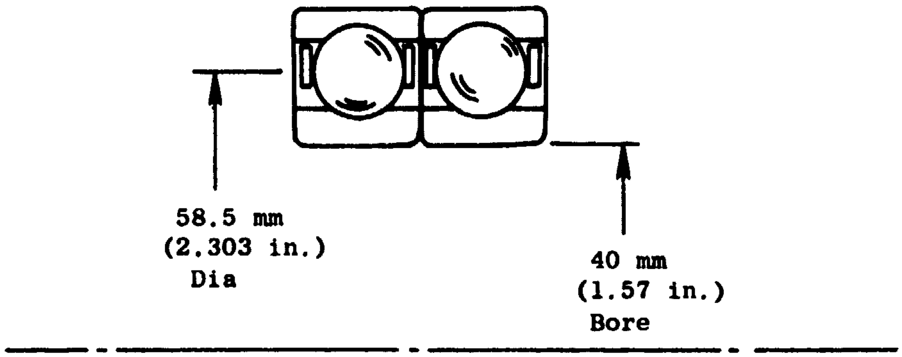
The forward axial dynamic stop configuration is shown in Figure 27. The stops consist of Belleville springs, keyed disks, and torsional jaws. Similar jaws, located on each end of the ball nut, engage the stops and drive one half of the disks with respect to their mating faces. This design dissipates energy for one full turn of the ball screw before hard axial stops engage and prevent further actuation. Upon disengagement of the jaws, return springs reposition the stops in their original axial location.

5.9 HIGH SPEED POWER SYSTEM

The high speed power system consists of the hydraulic motor, an LVDT feedback system coupled to the motor, a differential gear, and a no-back. The motor, differential gear, and no-back are shown in the actuation system cross section (Figure 1). This high speed power system has been designed and developed by Curtiss-Wright (Caldwell, NJ) under subcontract to the General Electric Company.

5.9.1 Hydraulic Motor

The hydraulic motor selected for the GE actuation system is an existing flight weight design currently used on the Lockheed L1011 aircraft for wing flap actuation. The hydraulic motor configuration and basic design characteristics are presented in Figure 28. Utilizing the maximum per blade twisting torque during actuation of 31,870 cm-N (2821 in.-lb) and working through the established gear ratios utilizing an estimated "worst" system efficiency, the motor must provide a maximum input torque of 18.13 m-N (160.5 in.-lb). The



- 12-12.7 mm (0.5 inch) Diameter Balls Per Row
- Cubic Mean Thrust Load = 55,026 N (12,371 lb)
- Calculated B_{10} Life/Required B_{10} Life*

  78 Hours/46 Hours

* Required Life Shown is for 48,000 Missions with Modulation on Approach.

Figure 26. Ball Screw Thrust Ball Bearing.

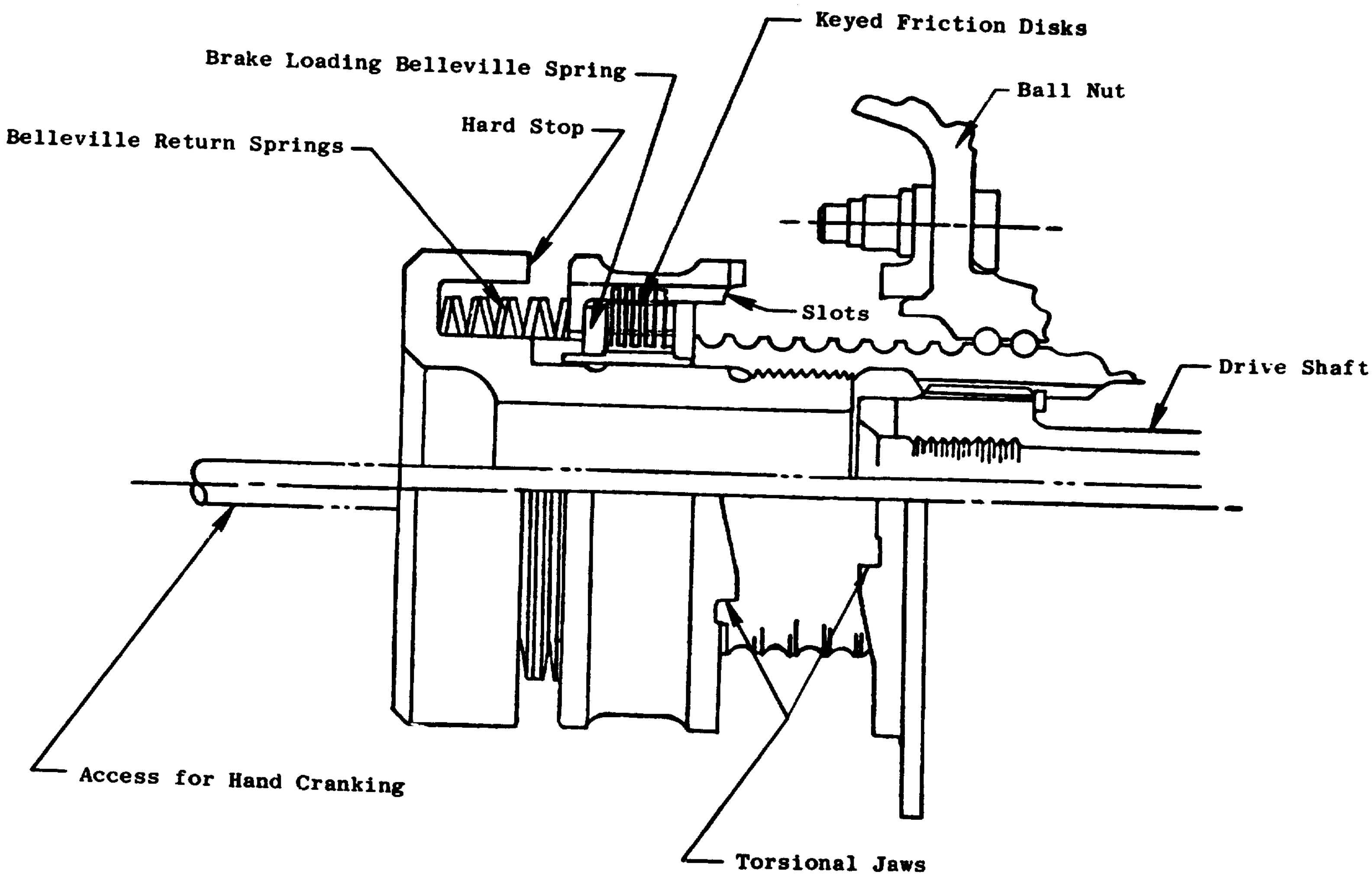
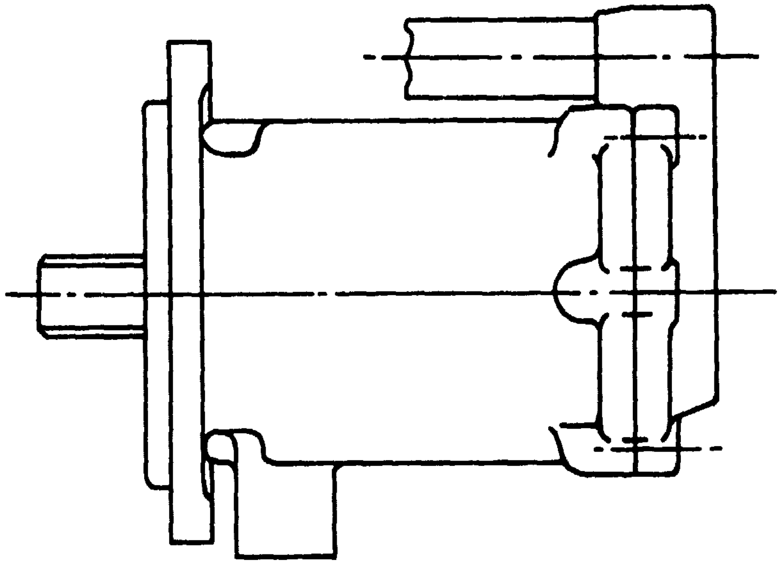


Figure 27. Axial Dynamic Stop (Forward).



- Vicker Model MS3-044-3 Fixed-Displacement 7.201 cm^3/rev (0.44 in.^{3}/rev) Multiple Piston (7) Hydraulic Motor
- Stall Torque = 20.34 Nm (180 in.-lb) at 2,379 N/cm^2 (3,450 psi)
- Torque = 19.89 Nm (176 in.-lb) at 10,000 rpm at 2,068 N/cm^2 (3,000 psi)
- Flow = 1,298 cm^3/sec (20.6 gpm) at 10,000 rpm

Figure 28. Hydraulic Motor.

hydraulic motor specification torque versus speed characteristics are shown in Figure 29.

The General Electric variable-pitch actuator hydraulic system (including the hydraulic motor, connecting piping, and electro-hydraulic servovalve) has been designed so the hydraulic polarity corresponds to the Hamilton Standard system.

5.9.2 LVDT Feedback Mechanism

The feedback mechanism, shown in Figure 30, uses two linear variable differential transformers (LVDT's) to sense fan blade angle. Rotation of the hydraulic motor output shaft is geared down and mechanically linked to an internally threaded shaft through a spur and worm gear combination. Rotation of the worm gear and mating sleeve shaft causes a mating externally threaded member to translate back and forth along its axis. Attached to each end of this translating member is the magnetic core of an LVDT.

Translation of the **magnetic core** of the LVDT produces an AC output voltage which is proportional to displacement. Thse transducers are constructed of one primary coil and two secondary coils. An alternating current is fed through the primary winding. The magnetic core couples the primary and secondary coils by conducting the alternating field inside the coils. When the core is in the center position, an equal portion of the core extends into each of the secondary coils and affects an equal coupling between the primary coil and each secondary coil. An alternating voltage of equal magnitude is induced in the secondary coils. With the secondary coils connected in series opposed, the output is close to zero. As the core is moved to either side, the coupling between the primary and one secondary coil is increased while the coupling between the primary and the other secondary is decreased. A larger alternating voltage then is induced in one secondary coil and the output voltage is the difference between the two voltages.

5.9.3 Differential and No-Back

The differential and no-back, shown in Figure 31, are the drive system connection between the hydraulic motor and the actuator assembly. This modular package attaches to the actuator and rotates at fan speed. Differential gearing connects the motor output shaft to the rotating no-back and results in a gear reduction of 5.44:1. Differential movement is transmitted through the no-back in either direction of rotation. The no-back, however, prevents imposed blade torque from back driving the system.

The no-back design is a ball-ramp-type configuration similar to the Curtiss-Wright design currently being used on the F-111 aircraft trailing edge flap system. The differential gear is similar to designs used previously in Curtiss-Wright turboprop pitch-change mechanisms.

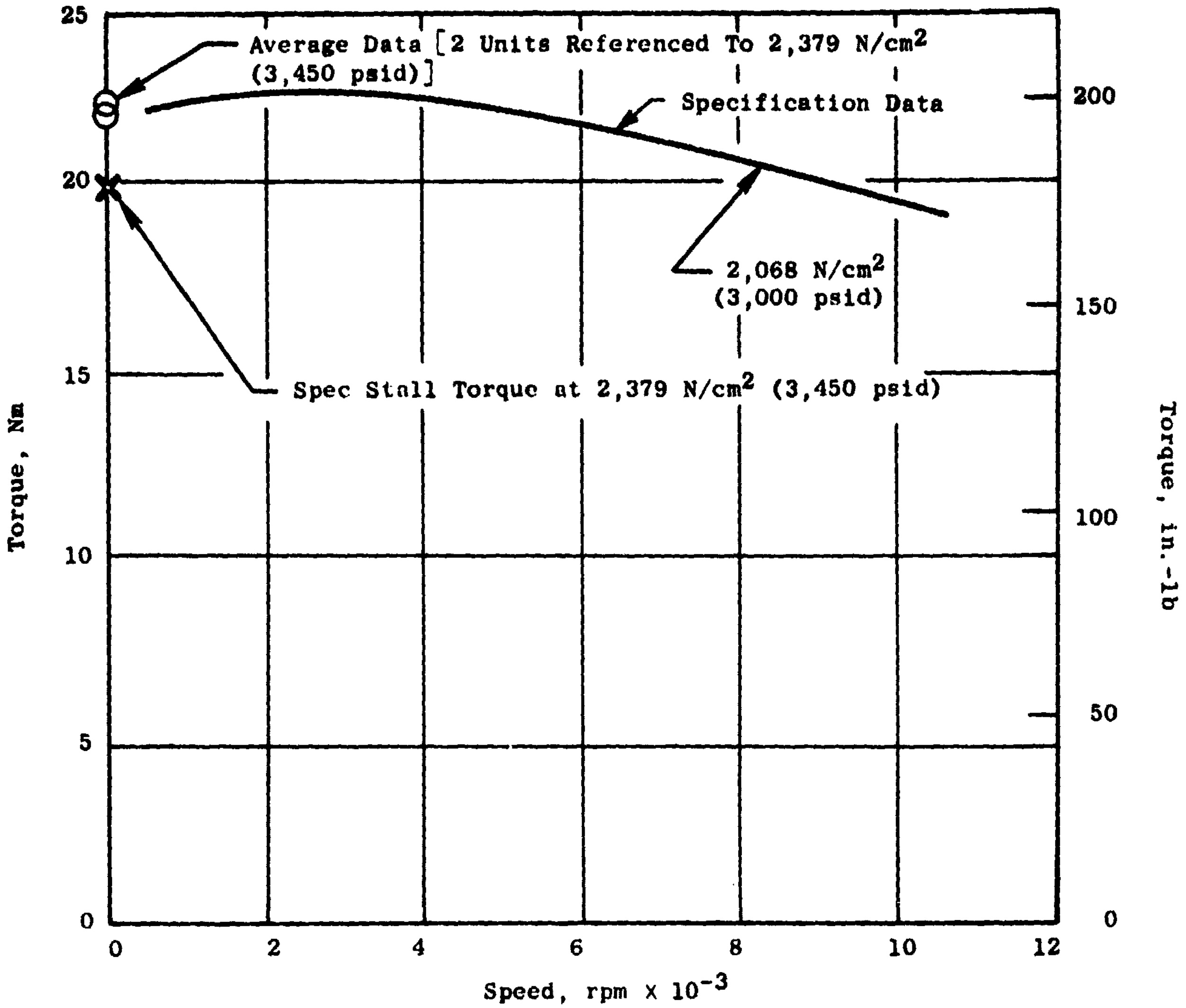


Figure 29. Hydraulic Motor Characteristics, Torque Versus Speed.

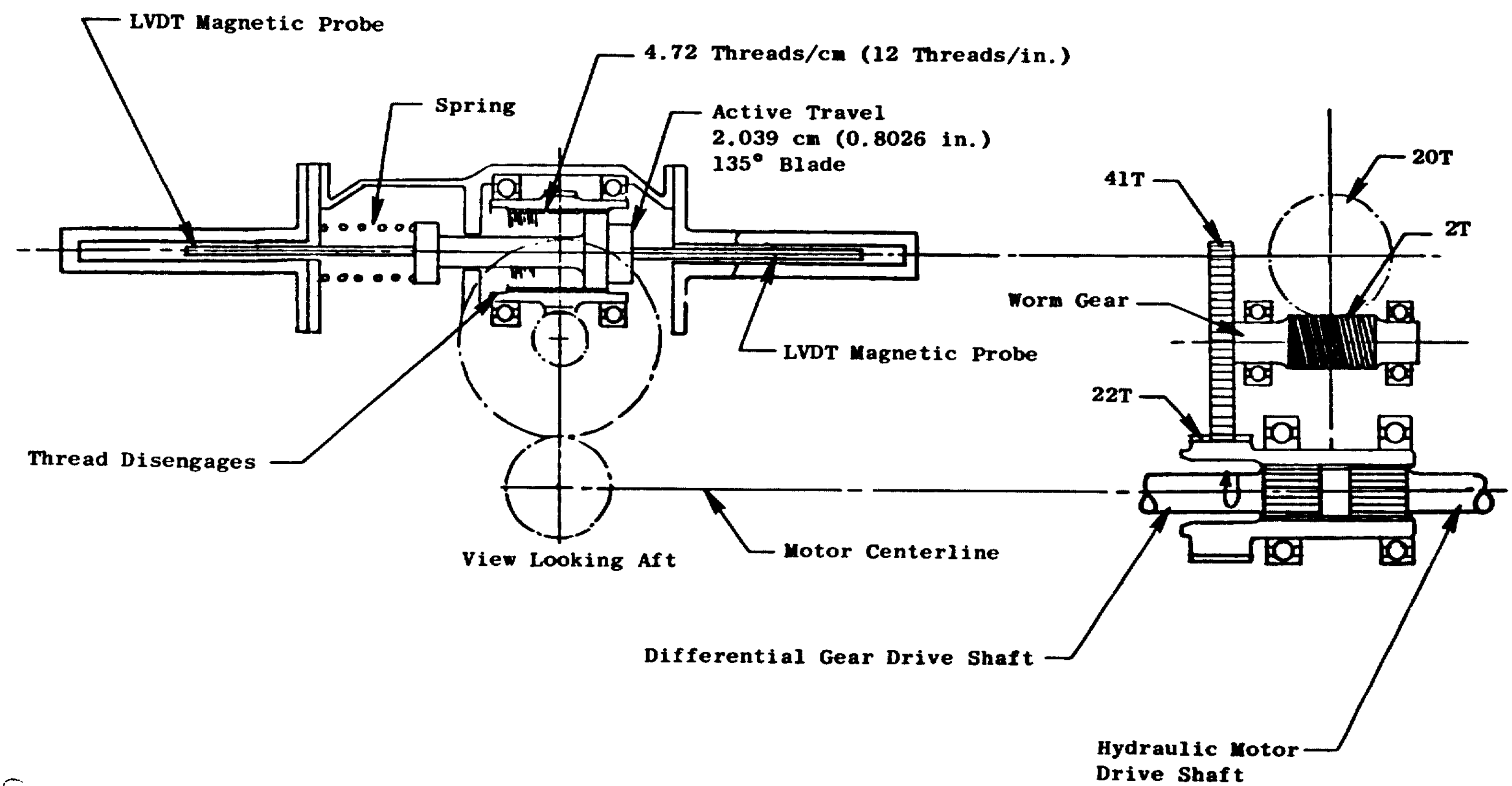


Figure 30. LVDT Drive Schematic.

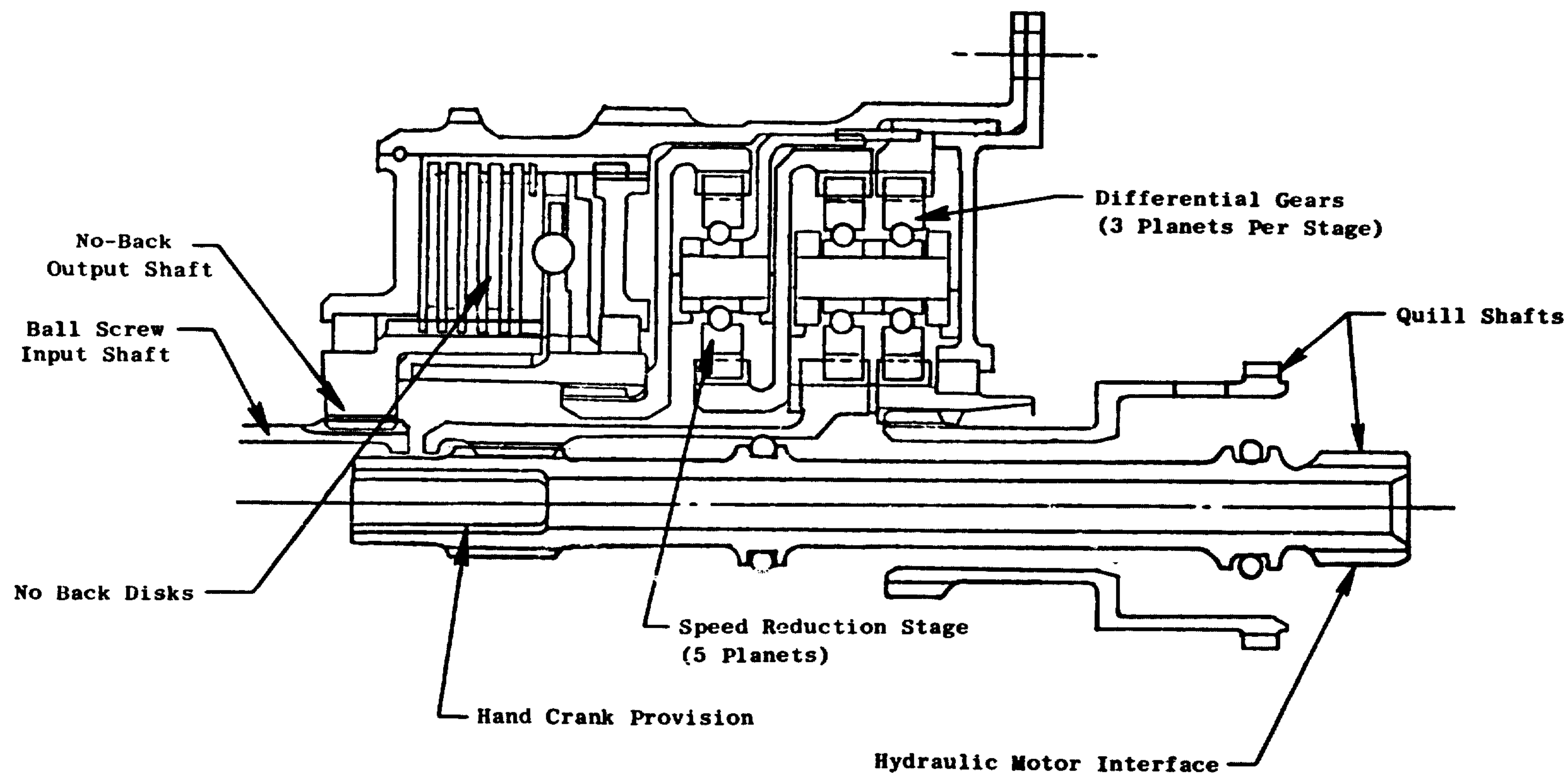


Figure 31. Differential and No-Back.

Common gears are used in all three stages of the differential gear. Coupling drive splines are designed to accommodate estimated misalignment during operation. The number of gear teeth and materials for the gears, gear bearings, and no-back disks are as follows:

N_T (SUN) = 27 N_T (PLANET) = 18 N_T (RING) = 63

Gear Material: H-11 through Hard (R_c 56-59)

Bearings Material: 52100

No-Back Disks: AISI 1095 with Sintered Bronze

As shown in Figure 31, provision is made along the engine centerline to hand crank the actuation system.

5.10 ELECTRO-HYDRAULIC SERVOVALVE

Hydraulic flow to the fan pitch and fan nozzle actuation is controlled by separate, four-way, electro-hydraulic, directional flow control valves mounted on a common manifold block which is mounted on the accessory gearbox. The valves control flow in response to a direct-current electrical signal from the digital control.

6.0 LUBRICATION SYSTEM

A schematic showing the actuator lubrication system is presented in Figure 32. As shown, oil from the engine lube system is supplied through the stationary hydraulic motor housing. A system of sleeves and dams centrifugally directs the lubricant to the critical areas of the actuator. The oil then is centrifuged outward and drains back to the engine sump along the outer actuator wall. All seals are designed so that no dynamic head of oil is present at the seal interfaces. The total actuator lubrication oil flow is 47.25 cm^3/sec (0.75 gal/min.).

The pinion and ring gears which cannot be lubricated from the engine lube system are to be coated per MIL-L-8937 (Everlube 620) which is an epoxy-based dry-film lubricant.

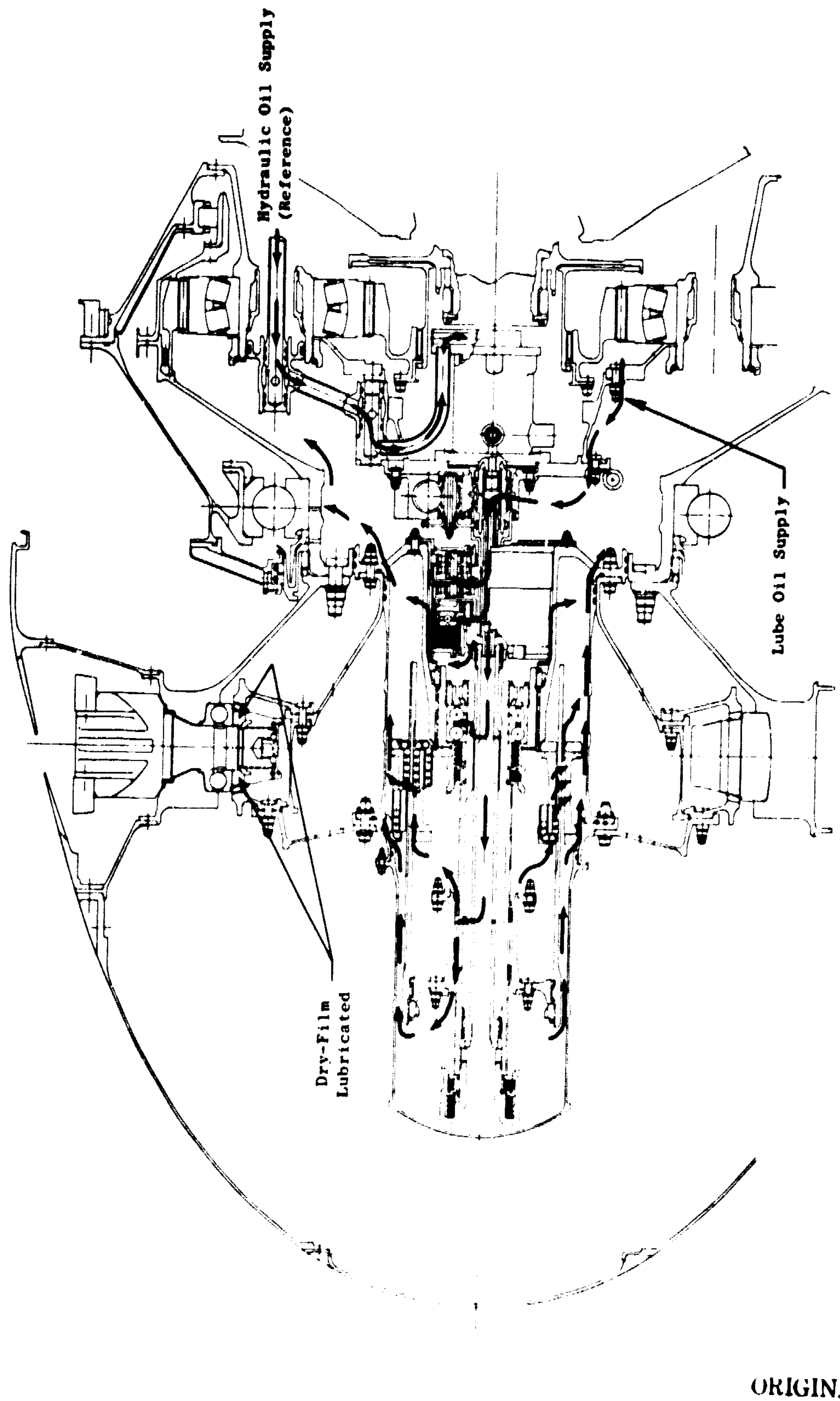


Figure 32. Actuator Lubrication System.

7.0 MAXIMUM BLADE PITCH ANGLE ACTUATION RANGE

The maximum blade angle range, assuming the actuator stroke is defined as the distance between the dynamic stops just starting to engage, is 130.6°, representing 14.99 cm (5.90 in.) of stroke. To prevent touching stops and to allow for tolerances only, a 14.48-cm (5.70-in.) stroke (126.1°) will be considered as the normal operating blade angle actuation range.

With the actuator set up to go from the nominal blade setting to reverse through stall, there is capability to go to -116.01° (open) and 10.13° (closed). By indexing pinion teeth (100.8°) when setting up to go to reverse through flat pitch, there is capability to go to 15.21° (open) and 110.93° (closed).

The actuation system is designed to permit an increase in the maximum open or closed positions for either reverse through flat pitch or reverse through stall simply by reindexing the blade pinions with respect to the ring gears.

The calculated mechanical blade angle error based on the system backlash is ±0.446°, which includes the items shown in Table IV. These calculations are based on worst-stackup conditions.

Table IV. Mechanical System Backlash in Terms of Fan Blade Angle.

	Mechanical Backlash
Ball Spline	±0.203°
Ball Screw	0.022°
Thrust Bearing	0.120°
No-back Play	0.084°
LVDT Drive Train	0.013°
Involute Splines	0.004°
	±0.446°

8.0 ACTUATION SYSTEM WEIGHT

The weight of the experimental engine actuation system is 69.05 Kg (152 lb) and is tabulated by various components in Table V. Weight of the hydraulic motor support, LVDT drive, servovalve, and no-back/differential gear assembly would be reduced for the flight-type configuration. The approximate weight of a flight-type system would be 62.2 Kg (137 lb). Further reductions in weight may be realized by optimizing the ball screw and ball spline stroke.

Table V. Experimental Engine Actuation System Weights.

Component	Weight	
	kg	pounds
Pinion Gears (18)	4.65	10.25
Actuator	46.72	103.00
Ring Gears (2)	15.84	34.92
Ball Spline and Translating Member	16.00	35.27
Ball Screw, Thrust Bearing, and Ball Nut	5.11	11.27
Drive Sleeve	3.28	7.22
Cover, Miscellaneous	6.49	14.32
Differential, No-Back, and Hydraulic Motor	7.25	15.98
Miscellaneous	10.43	23.00
Total	69.05	152.23

9.0 RELIABILITY

A detailed reliability analysis was conducted for the General Electric variable-pitch fan actuator system. During the initial phase of the study, a detailed failure modes and effects analysis was performed for each of the component systems, and failure consequences were established. The primary tool used to establish component failure rates was the General Electric Reliability Analysis computer Program (RAP) which is utilized for reliability analysis on all GE engine programs. This program is based on engine field service data for previous and currently operational GE engines. Generic failure rates, included in the program files, are modified to include effects of design differences from the model by inputing critical design criteria such as stresses, strain rates, deflections, etc. For most cases the analysis starts with each of the individual elements of a given component, such as gears, cams, etc., and compiles a system failure rate considering the interacting interfaces of the individual elements, rather than utilizing whole component failure rate predictions as is often done during preliminary studies. Also considered during the study were secondary damage effects to other elements or components which would result from the failure of a given member. Comparisons of the RAP study results with available failure rate data for similar components also are made (when possible) prior to establishing the final predicted failure rates.

The results of the failure rate analyses are presented in Table VI in terms of failure rates per million flight hours for: unscheduled component removal (UCR); in-flight shut-down (IFSD); in-flight power loss (IFPL); and, unscheduled engine removal (UER). Data presented are based on the flight duty cycle and take into consideration the intermittent operation of the actuator components.

Based on the failure rate data of Table VI, the predicted mean time between failure predictions is summarized as follows:

	MTBUCR (Hrs)	MTBIFSD (Hrs)	MTBIFPL (Hrs)	MTBUER (Hrs)
High Speed Drive Package	50,474	5,464,481	2,923,977	204,750
Imput Drive Mechanism	16,790	1,721,170	1,721,170	906,618
Actuation System	35,474	6,250,000	6,250,000	840,336
TOTAL SYSTEM	9,297	1,082,251	923,361	139,334

Table VI. Reliability Summary.

Part/Component	No. Parts	Calculation Method/Reference Data Source	Failure Rate/10^6 Hrs UCR	IFSD	IFPL	UER
High Speed Drive Package						
LVDT (Parallel Redundancy)	2	Manual RAP & F101 Augmentor Fuel Control LVDT's	1.267	0.029	0.115	1.014
Elect. Connector + Leads (Feedback)	2 + 4 Leads	Manual RAP & F101 Augmentor Fuel Control Connectors + Leads + L-188501-D13 Cannon Plug (Propeller).	1.480	0.015	0.030	1.184
EHV (Electro-Hydraulic Servovalve)	1	Manual RAP+F101A8 "Mogue" Servo.	6.220	0.015	0.062	0.622
Connector-Electrical Input (to EHV)	1 + 2	Manual RAP+F101A8 Augmentor Fuel Control Connectors+Leads.	1.072	0.011	0.022	0.858
Hydraulic Motor (Gear Type)	1	Vickers Hydraulic Motor P/N 24355K Flap on L1011, 2 Motors Per Actuator, No Failures. 671,000 A/C Hours, 90 A/C, 32 Months Operation	0.373	0.004	0.004	0.298
LVDT Screw Thread Drive Mechanism	2	Manual RAP Study + Sync.-Actuator C5-Thrust Rev.	1.000	0.001	0.001	0.800
Supply Tube to Hydr. Motor	1	Manual RAP Study + L-188 501-D13 Oil Tube Hub to Regulator	0.800	0.008	0.008	0.008
Support Cone (Hydr. Motor)	1	Manual RAP Study	2.200	---	---	---
Rotating Seal	1	Manual RAP Study	1.400	0.070	0.070	0.070
Thrust Bearings (Hydr. Motor Input)	2	Manual RAP Study + L-188 501-D13 Torque Cyl. Seal	4.000	0.030	0.030	0.030
			19.812	0.183	0.342	4.884
Input Drive Mechanism						
No-Back Assembly	1	Manual RAP Study	6.710	0.90	0.90	0.190
Differential Gearing	1	RAP Study of Hydromechanical Crossdrive Transmission XM2	21.100	0.211	0.211	0.633
Drive Shaft, Splines, Bearings	1 ea.	Manual RAP Study	28.000	0.230	0.230	0.230
Stator & Rotor Supports	3	Manual RAP Study	2.720	---	---	---
Rotating O-Rings	2	Manual RAP Study	1.030	0.050	0.050	0.050
			59.560	0.581	0.581	1.103
Actuation System						
Ball Spline Assy.	1	Manual RAP Study	18.550	0.020	0.020	0.450
Master Gears	2	Manual RAP Study	1.200	0.060	0.060	0.120
Ball Nut & Ball Screw	1	Manual RAP Incl. Design Parameter Calculations	1.500	---	---	---
Thrust Bearings	2	Manual RAP Study	4.000	0.030	0.030	0.540
O-Ring Seals	2	Manual RAP Study	2.200	0.050	0.050	---
Misc. Incl. End Cap & Ret. Ring	1 ea.	Manual RAP Study	0.740	---	---	0.080
			28.190	0.160	0.160	1.190
		TOTAL FOR SYSTEM	107.562	0.924	1.083	7.177

UCR - Unscheduled Component Removal
IFSD - In-Flight Shutdown
IFPL - In-Flight Power Loss
UER - Unscheduled Engine Removal

10.0 REFERENCES

1) "Quiet Clean Short-Haul Experimental Engine (QCSEE), Under-The-Wing (UTW) Final Design Report", NASA CR134847, June 1977, prepared by General Electric Company for NASA-Lewis Research Center.

2) "Quiet Clean Short-Haul Experimental Engine (QCSEE), Hamilton Standard Cam/Harmonic Drive Variable Pitch Fan Actuation System Detail Design Report", NASA CR134852, October 1977 prepared by Hamilton Standard Division of United Technologies Corp. for NASA-Lewis Research Center.

Appendix A

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An objective study has been made to select an optimum design for the Variable Pitch Fan actuator. A number of schemes were studied and systematically dropped along the way, leaving three basic systems which were evaluated in more depth. These systems include:

- Ball Spline Actuator
- Planetary Gear Actuator
- "Mini" Gear Actuator

Ball Spline Actuator

Using the technology (see Figure A-1) which has been successfully demonstrated during the Variable Pitch Fan and the Reverse Pitch Fan (VPF/RPF) programs, a second generation ball spline actuator has been studied.

The actuator, shown in Figure A-2, functions in the following manner. Bevel gears attached to each of 18 fan blades are rotated by the motion of two counteracting master gears. These master gears are rotated by a double acting helical ball spline driven by a rigid translating sleeve. A ball screw drives the translating sleeve through a stroke of four inces to achieve a blade rotation of 135 degrees. All thrust loads developed are close looped within the actuator mechanism.

Torsional stops at each end of the ball screw limit the actuator travel. Mechanical stops in the ball spline, although normally not engaged, are a backup system to limit the stroke. In this particular design the large master gears are easily reindexed to permit demonstrator testing through both flat pitch and stall pitch.

At maximum fan speed stop-to-stop actuation time is one second. A Linear Variable Displacement Transformer (LVDT) is used to provide the intelligence to the position feedback system.

The significant improvements of this second generation actuator over the VPF/RPF design are:

1. Two counter-rotating master gears are provided for redundancy by connecting the inner spline member to a second master gear instead of the fan disk.

2. A ball screw has replaced the hydraulic piston and transfer seal assembly. The screw is lighter, more reliable, and permits the ball spline to be packaged on a smaller diameter.

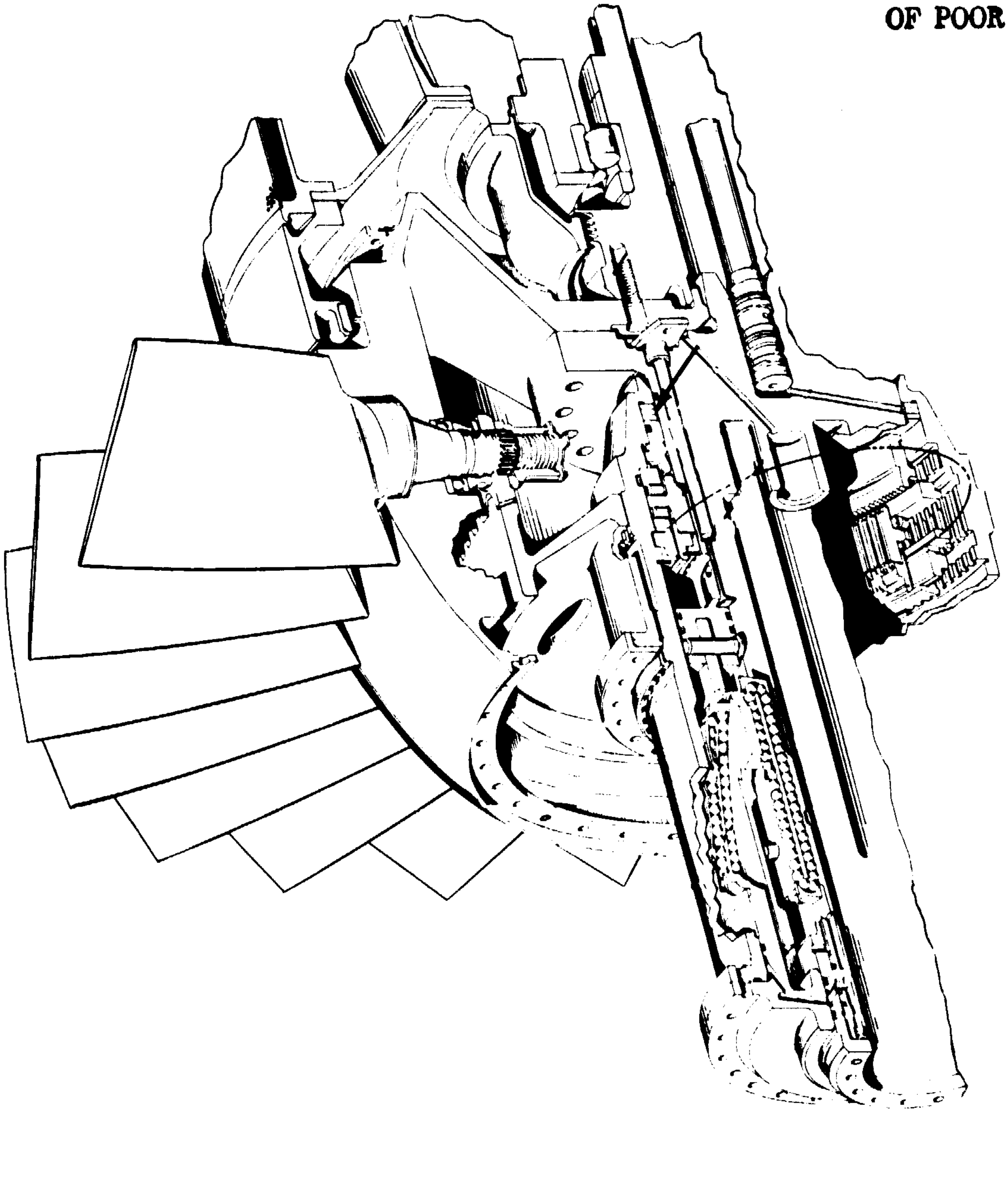

Figure A-1. Variable Pitch Fan Cross Section.

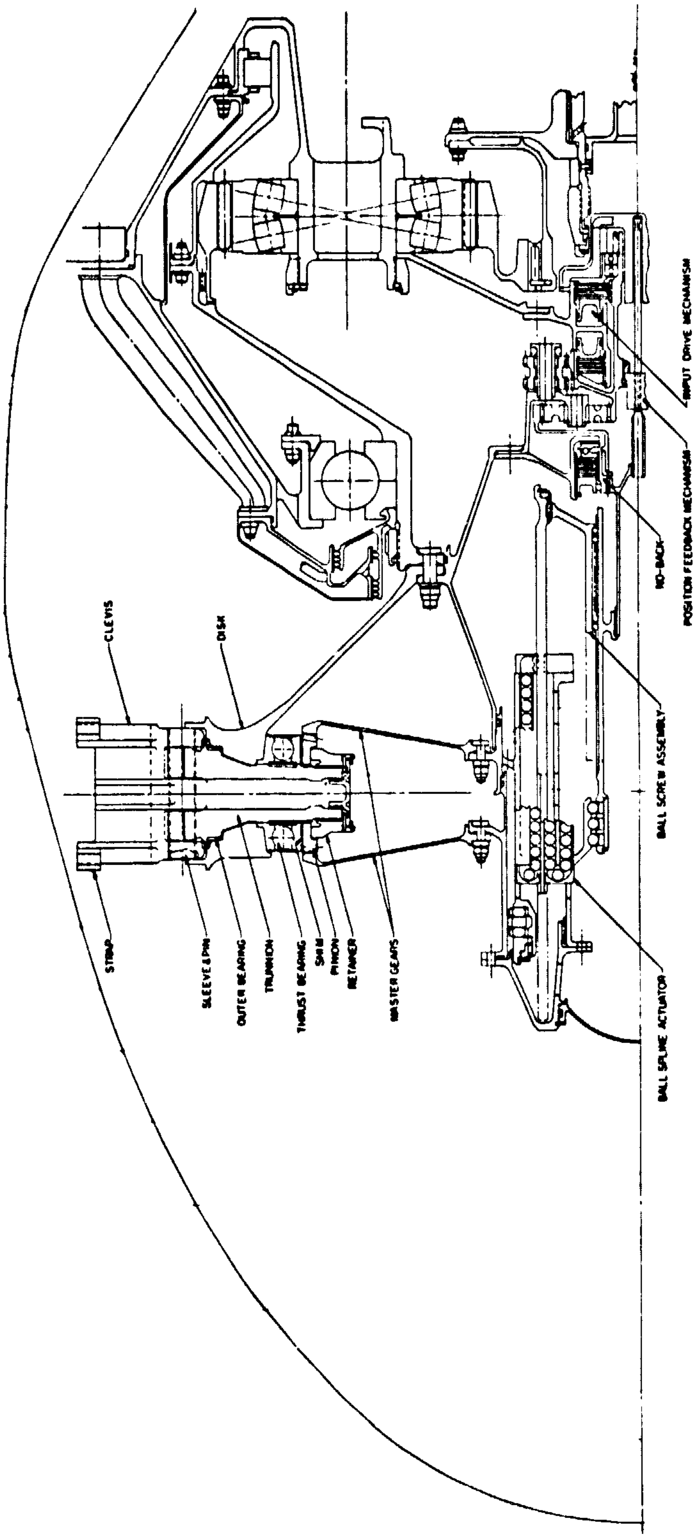

Figure A-2. Ball Spline Cross Section.

3. The pitch lock has been replaced by a "no-back". This device transmits input torque through the system, but prevents the net blade twisting torque from back-driving the actuator ball screw.

4. The VPF/RPF hydraulic actuator required a large high pressure pump, drive system, heavy piping, and a hydraulic transfer package. The QCSEE actuator input torque is supplied by the engine fan shaft. Two lightweight methods of extracting torque from the fan drive shaft have been studied and encompass the use of a differential gear controlled by either a system of disk brakes or a combination differential/lube pump.

The features of this system are:

1. Design is based on a demonstrated system which has proven trouble free.

2. High reliability based on simple rugged design.

3. Elements subject to wear are balls which are easy and inexpensive to replace.

4. The rolling action of the ball spline and screw offers a very high efficiency system which does not require additional gearing to reduce torque levels to the "no-back" and differential gearing.

5. The actuator design lends itself to modular assembly and disassembly and allows easy access to differential gearing.

6. Recirculating engine oil rather than grease packing is used for lubrication.

7. Overload torques associated with bird impact requirements are satisfied.

Planetary Gear Actuator System

A planetary gear actuator has also been studied and is shown in Figure A-3. Bevel gears attached to each of 18 blades are driven by two counteracting master bevel gears. These two gears are driven from the output ring gears of a 109:1 gear ratio compound planetary gearbox. The input to the reduction gear is the carrier which also supports the 12 compound planet gears. The high gear ratio is accomplished by closely matching the output ring gear diameters to each other.

The planet gears are supported between the carried walls which, along with the extended ring gear outside shell, form a greased cavity. The required grease seals are located out of the centrifugal field. A thrust bearing is located in this cavity to counteract the master bevel gear thrust loads.

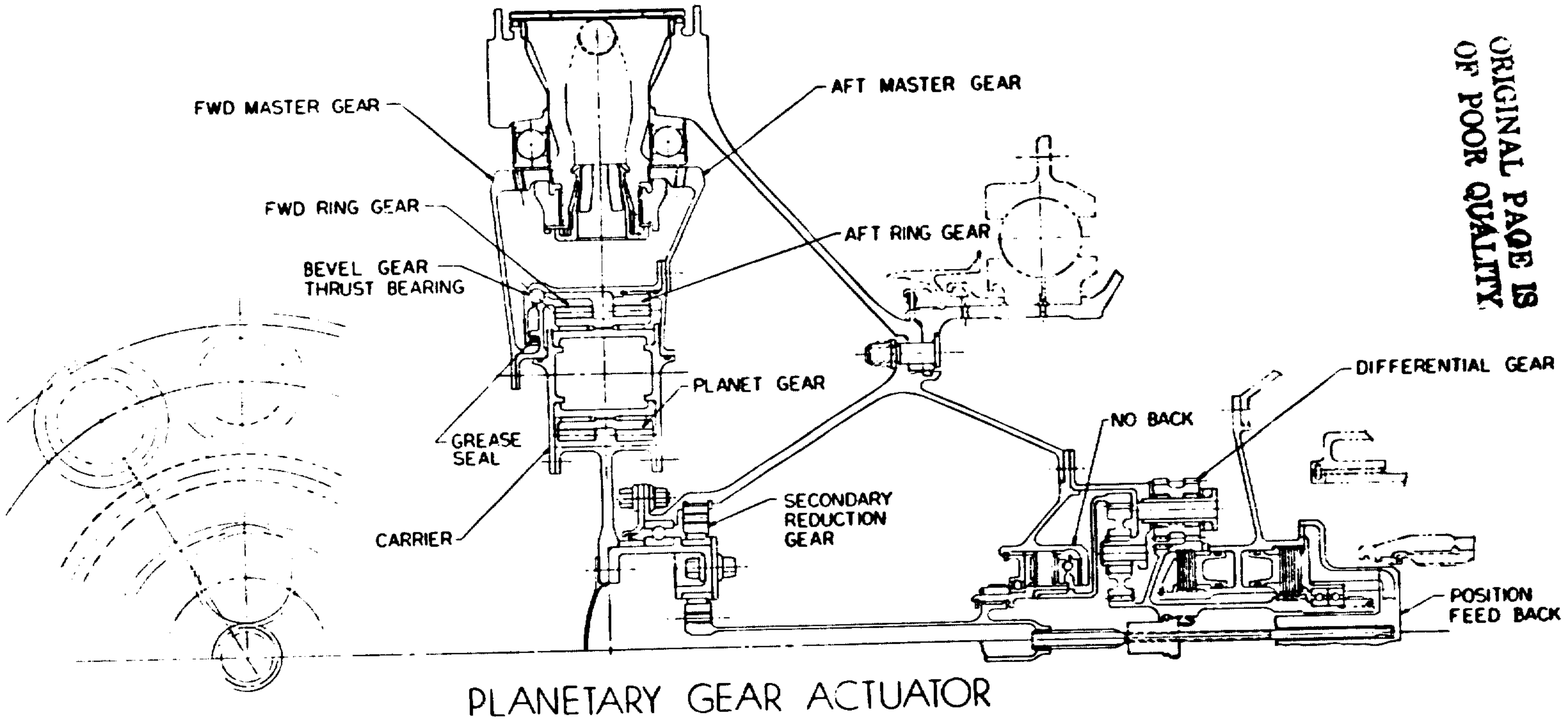

Figure A-3. Planetary Gear Cross Section.

A secondary reduction gear is mounted to the carried of the compound planetary gear to further reduce the actuator torq· levels.

The mechanism is driven by differential gearing and the direction of blade rotation is controlled by a system of brakes. A "no-back" is provided between the differential and the actuator gearing the maintain blade angle settings.

Position of feedback is provided by using the LVDT mounted to the brack housing which relates the position of a sliding rod to the blade tangent angle.

The Features of this system are:

1. Low system weight due to high gear ratio planetary gear set.

2. High torque reduction obtained by a relatively small number of parts.

3. Large gear tooth sections allowing large root radii, therefore, low stress concentration.

4. Readily accessible for inspection and servicing after removing engine spinner.

5. Low torque requirements for the no-back differential gear and the disk brakes.

6. Planetary gear actuator can be assembled and removed as a module from the engine.

7. Differential gear can be removed as a module without disturbing hydraulic and electrical connectors.

"Mini" Gear Actuator

The "mini" gear actuator is shown in Figure A-1. This concept features individual high gear ratio drives (mini-gearboxes) secured to each of the 18 fan blades. These gear drives are synchronized to each other by bevel gears connecting all 18 "mini" gearboxes. This synchronizing arrangement reduces the overall actuation torques by eliminating the need for a large master gear, but does cause adjacent bevel gears to rotate in opposite directions. To provide common rotation for the fan blades, each adjacent "mini" gearbox is kinematically arranged so the same output rotation can be obtained with opposite input rotation.

The "mini" gearbox utilizes compound planetary gearing where one stage is modified in each adjacent gearbox to obtain a common output rotation. High gear ratios are obtained by designing the two ring gears as close in diameter as possible. With the kinematic arrangement shown many ratio combinations can be established and 85:1 was used in this study.

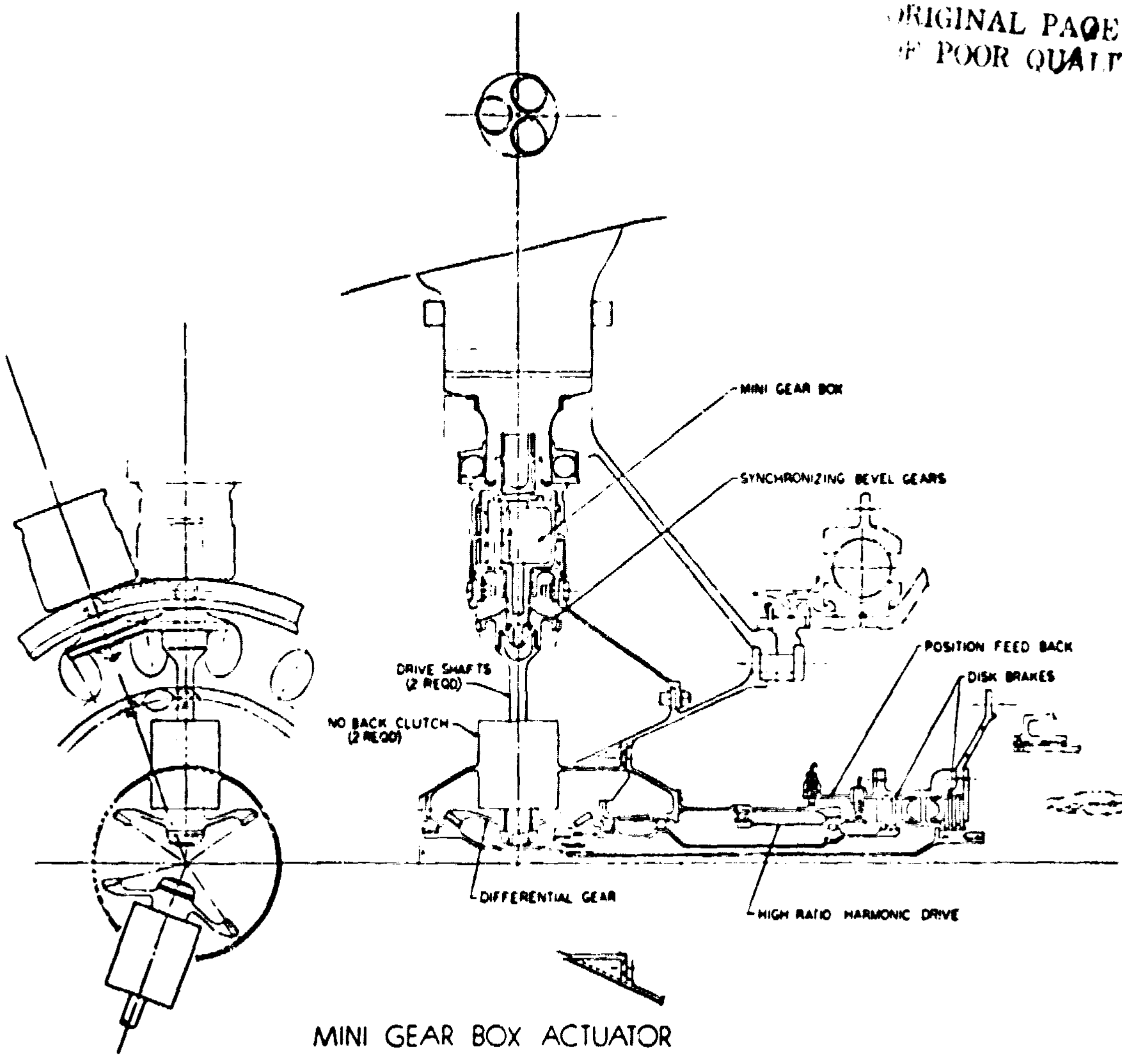

Figure A-4. Mini-Gear Cross Section.

The centrifugal load imposed on the individual planets is reacted by thrust washers at each planet and by a central spherical seat bearing. Each unit is grease packed and the required rotating seal is out of the centrifugal field.

Rotation is supplied to the synchronized "mini" gearboxes by two drive shafts attached to appropriate bevel gears. These shafts are driven by a bevel gear differential through a "no-back" which will maintain blade angle settings. The "no-back" can be a spring type or Sprag clutch. Holding torques will be low because of the high gear ratio in the "mini" gearboxes.

The output direction of the differential gearing is determined by stopping either of the input bevel gears. This can be accomplished by a system of disk brakes as shown in Figure A-4, or a combination lube/actuator pump.

Position feedback isn provided by using a high ratio harmonic drive whose output gives a phase shift signal between the fan shaft and the differential gearing during actuation.

The features of this actuation concept are:

1. Low overall torque levels because of high gear reduction at the blade hub.

2. By adding only one more bevel gear to the right angle bevel gear train, a differential function can be provided.

<u>Actuation Configurations</u>

In comparing the three systems studied, the product end use was weighed strongly in the selection of the prime candidate. The selected design must be suitable for airline use considering such items as reliability, ease of maintenance, weight and production unit cost.

From the standpoint of reliability, there is an advantage of the ball spline actuator over the planetary gear based on its concept simplicity. The rolling action of a ball and the precision which can be obtained when compared with the sliding (more wear) action of a gear makes the ball spline a more reliable design. Great care must be taken during manufacture and assembly of a planetary gear to prevent overloading of one planet and thus leading to premature failure. Balls are easily replaced when servicing is require. The "mini" gear has a very high individual part count and cannot be considered as reliable as the ball spline or the planetary gear.

Maintenance of the ball spline will be less costly than any gear system. The cost of ball replacement is low compared to the replacement of any gearing.

It was originally thought that the ball spline design would be heavy and that the "mini" gear concept would represent a lightweight system. This has not been the case. The weight of each "mini" gear is approximately 2.95

pounds or, when considering 18 units, is 53.1 pounds for the "mini" gears alone. It has been found that all three systems are competitive from a weight standpoint. The ball spline became competitive when the ball screw replaced the hydraulic piston for translation of the central member.

Production costs of the ball spline and the planetary gear will be competitive, but the production costs of the "mini" gear will be higher based on its complexity.

Also to be considered in this program is development risk. Here the ball spline has a definite advantage. The design has evolved from a developed technology and presents a minimum risk development program.

Based on this study the ball spline actuator is recommended for the QCSEE Variable Pitch Fan System.

System Trade Summary

Table A-1 shows the results of the trade study that led to the selection of the ball spline actuator for the Variable Pitch Fan System. This is a result of a detailed study of three actuation systems. The selection of the prime candidate was made on the basis of the items of evaluation shown in the General Electric Statement of Work which encompass:

1. Suitability for Airlines use
2. Reliability
3. Weight
4. Development Risk
5. Development Cost
6. Maintainability
7. Production Cost
8. Development Flexibility.

A ranking of 1 to 3 has been used in Table A-1 where 1 represents a more desirable and 3 a less desirable design. The column identified as "Results" is the overall rating of the particular design feature.

Table A-1. Tabulation of Tradeoff Study Results.

Component	Suitability for Airline Use	Reliability	Weight	Development Risk	Development Cost	Maintain-ability	Production Cost	Development Flexibility	Bird Impact	Results
Actuation Mechanism										
• Ball Screw	1	1	1	1	1	1	1	1	1	1
• Planetary Gear Actuator	1	2	1	2	2	1	1	1	2	2
• Mini Gear Actuator	3	3	2	3	3	3	2	3	2	3